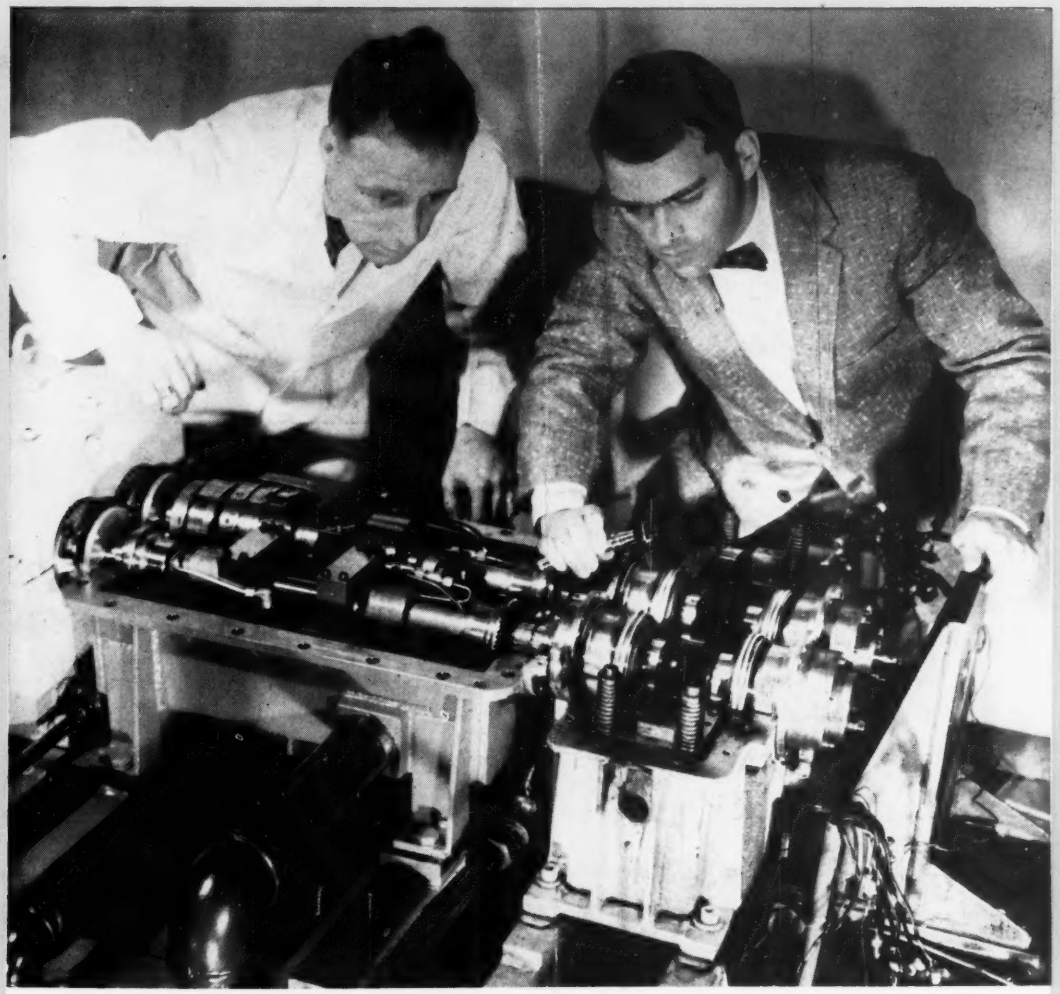
Vol. 15, No. 3 March, 1959

LUBRICATION ENGINEERING

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE



High-Temperature Gear Lube Tests (see page 98)

JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

FARVAL—*Studies in Centralized Lubrication*
No. 218

Farval lubrication first choice for Hill-Acme Upsetter

On this 1¼-inch Hill-Acme Upsetter, Farval's timing cycle can easily be changed to conform with the service of the machine. Amount of lubricant delivered to each bearing is also adjustable.

BUILDING peak performance into their machines is a point-of-pride with Hill-Acme. That's why you'll find Farval lubrication systems on H-A machines such as this upsetter.

When the going's rugged, with shock, vibration and wear trying to take their toll, Farval continuously guards 59 vital bearings . . . continuously delivers measured amounts of lubricant at regular intervals.

More and more, machine designers, engineers and production men are turning to Farval for the kind of lubrication that spells uninterrupted, maintenance-free production. Let us tell you more. Write for Bulletin 26-S. The Farval Corporation, 3267 East 80th Street, Cleveland 4, Ohio.

KEYS TO ADEQUATE LUBRICATION

Wherever you see the sign of Farval—familiar valve manifolds, dual lubricant lines and central pumping station—you know a machine is being properly lubricated.

Affiliate of The Cleveland Worm & Gear Company, Industrial Worm Gearing. In Canada: Peacock Brothers Limited.

Volume 15, No. 3
March, 1959

Lubrication Engineering

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

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14th ASLE Meeting To Bring Experts To Buffalo in April

The 14th Annual Meeting and Lubrication Exhibit of the American Society of Lubrication Engineers will be held in Buffalo, N.Y. at the Hotel Statler on April 21, 22, and 23.

More than 2000 top-industrial executives and lubrication engineers, coming from all over the world, are expected to hear 46 lubrication experts scheduled to speak during the three-day event.

Outstanding topics include "New Concepts in EP Greases," "Progress in Roller Bearing Lubrication," "Infrared Hot Box Detection," and a specialized paper on "Lubrication Problems in the Cement Industry." Many subjects to be covered are aimed at correcting common problems existing in industrial plants and improving production by minimizing downtime due to premature mechanical failure. Complete programs will be mailed on or about March 15.

Among the outstanding speakers who will address the Society this year is R. K. Sedgwick of Kearney & Trecker whose paper "Friction, Stiction and Fiction" promises to dispell some of the myths surrounding lubrication.

Other authors include B. L. Mordike of the University of Cambridge, England; A. Cameron, University of London, England; and N. Tipei of the Institute of Applied Mechanics, Bucharest, Romania.

In addition to the wide range of subjects to be covered in the meetings, the Society is also sponsoring its annual Lubrication Exhibit in the Statler Showroom on the mezzanine floor of the Hotel Statler. Here members and guests will see every conceivable device for modern lubrication practice, as well as the latest developments in lubricants for industry.

The annual Short Course in Lubrication Engineering will be held on these same three days in the Buffalo Room of the Hotel Statler. Highlight of this course in lubrication will be a tour of the Lackawanna Works of the Bethlehem Steel Company with particular emphasis on lubrication applications.

Tour of the Falls, Chic Fashion Show In Ladies Program

While providing plenty of entertainment, this year's Ladies Program leaves enough free time for wives of those attending the Annual Meeting to strike out for themselves to see the sights of the Buffalo area.

Highlight of this year's program, as planned by Mrs. E. L. Markle and her able committee, is an all-day tour of Niagara Falls—the Canadian as well as the American side. Foreign-born guests must present naturalization papers to customs inspectors. Those planning this trip must register on Monday, April 20.

A luncheon and fashion show will be an additional feature of the conclave. Advance reservations are necessary.

The complete Ladies Program is as follows:

Monday, April 20, 1959

12:00 to 5 p.m. Registration. Mezzanine Lounge, Hotel Statler. Fee $5.00.

7:30 p.m. Welcoming party given by Buffalo Section in Buffalo Room, Hotel Buffalo.

Tuesday, April 21, 1959

9 a.m. to 5 p.m. Registration.

Coffee time. The Club Room, through the courtesy of the Manzel Div., Houdaille Industries, will be available for the ASLE ladies from 8 a.m. to 5 p.m. Tuesday and Wednesday.

9 a.m. Departure of bus tour of Buffalo.

12 noon to 3 p.m. Luncheon at Mary Seton Room followed by fashion show at Kleinhan's Music Hall.

3 p.m. Tour of Kleinhan's Music Hall.

3:30 p.m. Return to hotel.

Wednesday, April 22, 1959

9 a.m. to 10 a.m. Coffee time.

10:15 a.m. Departure time for all-day Niagara Panorama Tour. Busses will return to hotel sometime in mid-afternoon. Bus transportation will carry guests across Peace Bridge into Canada and to interesting points in Niagara Falls, including a stop for shopping. Lunch will be served in the Rainbow Room of the Sheraton Brock Hotel, Niagara Falls, Ont.

Thursday, April 23, 1959—No organized activity.

Shuffle Off to Buffalo

Buffalo has often been described as "the city of trees" and the "city of good neighbors." In addition to beauty and friendliness, however, the city has become one of the leading commercial and industrial centers of the country. Airplane manufacturers, steel plants, great grain elevators, and the activity of the world's largest inland harbor, combine to make Buffalo a most diversified industrial center.

The second city in population in New York State, it is located almost equidistant from New York City, Boston, Philadelphia, Baltimore and Chicago.

Buffalo proudly claims one of the finest music halls in the world, Kleinhans Music Hall. It is the home of the Buffalo Philharmonic Orchestra and the scene of operas, concerts, lectures, and a host of other cultural events.

One of the wonders of the world is the awe-inspiring spectacle of Niagara Falls. Visitors from all over the world are drawn to Buffalo to view the beauty of this phenomenon of nature. A reminder of America's past is Fort Niagara. Located a few miles from the city limits, Fort Niagara takes the visitor back to 18th century Revolutionary War days.

Scores of theatres, restaurants, night clubs and open-air dance pavilions give the city a spirit of bouyant fun which is felt by the visitor immediately on arrival.

Also of interest are the sports events held in the Civic Stadium. Many sports events, featuring midget and stock car races, are presented regularly.

Whatever your interests, you're sure to enjoy Buffalo.

More than two million annual visitors flock to Buffalo to view the wonders of Niagara Falls.

Buffalo's Main Street boasts many forms of entertainment promising gay, sparkling fun for the visitor.

Unique triple-lip National Syntech® seals bearings "underground" in rugged disc harrow application

Low cost, easily installed
Excludes dirt, mud and water
Ideal for permanently lubricated bearings
Effective after 4,000 test hours

The Triple-Lip Syntech seal, pioneered by National, provides a new standard of bearing protection for equipment operating in severe dust, dirt, mud and water conditions.

The seal is rugged and extremely simple in design. It consists of three identical synthetic rubber sealing members bonded to metal washers and enclosed in a rigid steel outer case. Use of the "straight" Syntech sections keeps torque low, simplifies flush lubrication, and permits the seal to accept a high degree of runout and misalignment where needed.

For complete information, call your National Applications Engineer or write direct

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Tour, Film, Talks Set in Lube Course

The tentative program for the 1959 short course in lubrication engineering has been set. The short course is a regular part of the ASLE Annual Meeting, this year's session being held in the Buffalo Room of Buffalo New York's Hotel Statler. The 1959 program features a new approach to the annual course.

Sponsored by the ASLE Education Committee and its chairman, M. E. Dougherty of Acheson Colloids Co., the short course will be held concurrently with the technical sessions scheduled for the April 21, 22 and 23 Annual Meeting.

The program will be launched with an introduction by Theodor Ranov of the University of Buffalo's School of Engineering. Professor Ranov will explain the purpose of the course and will summarize the history and science of lubrication.

One of the highlights of the course will be a film on basic principles of lubrication which will be shown free of charge to lubrication course members and to engineering students in accredited colleges. Another high point of the program will be the question and answer panel which will close the course at noon of Thursday, April 23. A tour of Bethlehem Steel's Lackawanna Works is also planned. Lubrication course students will be permitted to attend any Thursday afternoon technical session.

The topics to be explained and discussed are: Thick Film Lubrication, Boundary Lubrication, Properties of Lubricants, Application of Lubricants and Maintenance of Lubricating Equipment, Solid Lubricants, Gears and Gear Lubrication, Bearings and Bearing Problems, Bearings—Maintenance and Lubrication, Seals and Packings, and Plant Lubrication and Organizing a Plant Lubrication Program.

A complete listing of the program, as planned to date, appears on page 94 of this issue.

Future Jet Lubes May Be Combined With Fluoroalcohols

Highly fluorinated alcohols may be the key to lubricants satisfactory for use at the elevated temperatures anticipated in future military jet engines, Du Pont scientists say.

Prepared from tetrafluoroethylene and methyl alcohol by telomerization, the fluoroalcohols can be reacted with selected di- or polybasic acids to form esters which appear to have the required oxidation, thermal, and hydrolytic stability for high temperature operation. Most promising compounds found so far, said Du Pont's E. C. Ballard and E. E. Sommers at the annual meeting of the Society of Automotive Engineers, are a fluoroalkyl camphorate and a fluoroalkyl pyromellitate.

Oxidation tests on these fluoroesters, the Du Pont chemists said, indicate the camphorate can be used at bulk oil temperatures up to at least 400F., while the fluoroalkyl pyromellitate extends the useable temperature range another 100 degrees.

Ballard and Sommers reported that a jet engine in which fluoroalkyl camphorate was used as a lubricant was found to be "very clean" after a 100-hour run at a bulk oil temperature of 300F. Additional tests at higher bulk oil temperatures will be required to establish the maximum performance limit of this lubricant, they said.

Although the fluoroester lubricants do not possess the excellent viscosity and low temperature properties desired in the tentative Military Specification MIL-L-9236, the Du Pont chemists indicated the problem might not be an insurmountable one.

Physicist Merchant Slated for 1959 National Award

The awards committee of the American Society of Lubrication Engineers has announced that Dr. M. E. Merchant has been named the recipient of the 1959 ASLE National Award. Dr. Merchant, who is Senior Research Physicist for the Cincinnati Milling Machine Company, will be presented the award at the 1959 Annual Meeting to be held in Buffalo, N. Y. on April 21, 22, 23.

The national award is the highest honor bestowed upon an individual by ASLE. It symbolizes an honorary life membership in the Society and is awarded in recognition of an outstanding contribution to the field of lubrication.

Dr. Merchant's many important contributions in the field of basic research in friction and lubrication qualify him as the outstanding man-of-the-year in this field. His work deals primarily with the chemistry of reactions occurring in metal cutting and metal cutting fluids. He has written and presented many valuable papers on these subjects.

Last year's ASLE National Award was received by Earle A. Ryder, well known consulting engineer.

M. M. Gurgo

M. M. Gurgo Named to Board

M. M. Gurgo has been elected to the Board of Directors of ASLE to fill the unexpired term of B. G. Symon, International Lubricants Corp., who recently resigned this office. Mr. Gurgo is associated with the ESSO Standard Oil Company of Pittsburgh, Pa. and will serve as industrial director until April 1960.

ASME Spotlights Use of Engineers

Consideration of more effective use of engineers by companies which employ them will highlight the topics to be discussed at the fourth Design Engineering Conference, sponsored by the machine design division of the American Society of Mechanical Engineers, in Philadelphia in May.

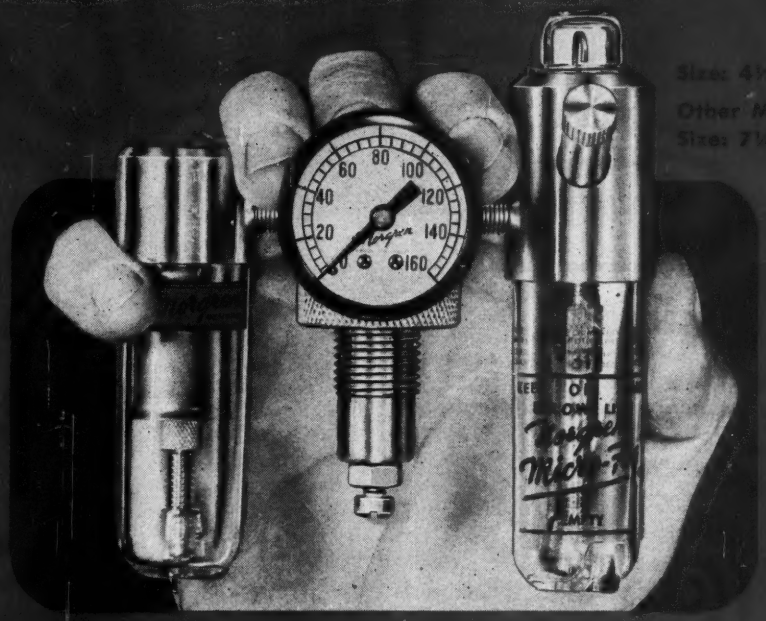
The conference is held concurrently with the Design Engineering Show. Both events are scheduled for Convention Hall, Philadelphia, May 25 to 28, inclusive. Conference sessions will take place in the mornings and the show during afternoons.

There will be two conference sessions dealing with the more efficient use of engineering talents and skills. The first, on the opening day, will consider the philosophy of engineering design in foreign countries. This will be a panel session and the discussion will center on the freedom of designers to design as they believe, as contrasted with design which is influenced by sales, marketing or consumer motivation.

Another session, on the last day, will discuss the efficient organization of a design department from the viewpoint of both the company for efficient operation, and the engineer for proper use of his talents.

On the second and third days of the conference, there will be three concurrent sessions dealing with choice of materials in design, mechanical aspects of design, and power and control.

Where Space is Tight...



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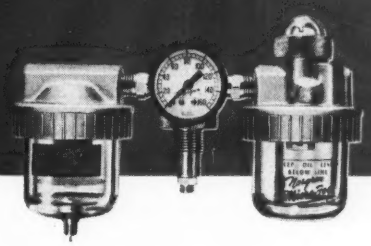

Models Available for Air Flows from .05 to 20 cfm.

These new Norgren MICRO-FOG units: **1.** Provide highly efficient compressed air processing for air tools and other air operated devices. **2.** Provide efficient lubrication for bearings and machine tool components. Maximum operating pressure 150 psi, maximum temperature 120° F.

AIR LINE FILTERS

Two new air line filters provide efficient removal of liquids and solids from the air line, preventing damage to connected air equipment.

PRESSURE REGULATOR

A new small, compact pressure regulator is used for both series of Lubro-Control Units. It provides accurate, trouble-free regulation of the air pressure. Adjustable over a pressure range of 5 to 100 psi.

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New MICRO-FOG Lubricators provide efficient and adjustable lubrication for bearings, air tools, air cylinders and other air operated devices.

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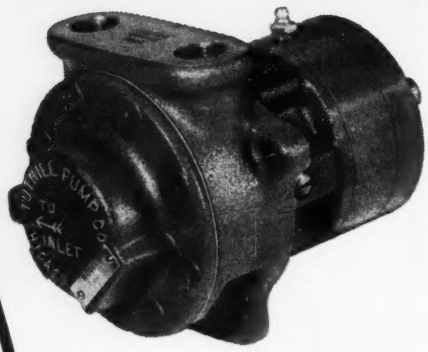
TUTHILL automatic reversing pumps

- **Positive reversing action**
- **Require no valves**
- **Port positions remain constant**



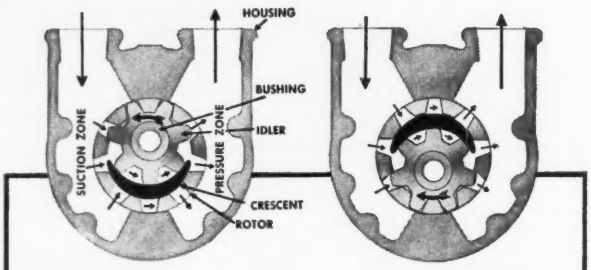
The model RCK above is typical of Tuthill's complete line of automatic reversing pumps . . . which use the time tested operating principle at the right to provide instantaneous, positive reversing action without the use of valves. The port positions remain constant regardless of the direction of shaft rotation. And all pumps provide uniformly high efficiency in both flow directions.

The automatic reversing design was developed by Tuthill for applications where the pump must be driven from a reversing shaft, or where machinery must be shipped without knowing the ultimate direction of the driving unit. These pumps have been enthusiastically accepted by designers and have proven their dependability in thousands of demanding applications such as large air compressors and machine tools.

375 Models

A complete selection of 375 models is provided with capacities from ⅓ to 200 gpm; for pressures to 400 psi; and speeds to 1800 rpm. Included are a complete assortment of stripped models specially developed for incorporation into manufactured products.

A 12-page catalog, No. 105, gives complete information on all Tuthill automatic reversing pumps. Write today for your copy. Or, if you desire, send drawings so that Tuthill's engineers can show you how the Model R reversing pump can be built directly into your product.



COUNTER-CLOCKWISE ROTATION

CLOCKWISE ROTATION

THE PUMPING PRINCIPLE

Tuthill automatic reversing pumps are based on the use of a rotor, idler gear and a crescent shaped partition which is integral with a moving part called the Idler Carrier.

Figure 1 shows how power is applied in counter-clockwise rotation to the rotor and transmitted to the idler gear with which it meshes. The space between the outside diameter of the idler and the inside diameter of the rotor is sealed by the crescent. When the pump is started there is an increase in volume as the teeth come out of mesh. This creates a partial vacuum, drawing the liquid into the pump through the suction port. The liquid fills the spaces between the teeth of the idler and rotor and is carried past the crescent partition to the pressure side of the pump. When the teeth mesh on the pressure side, the liquid is forced from the spaces and out through the discharge port.

When the shaft rotation changes from counter-clockwise to clockwise, the idler carrier (including the idler gear and crescent) automatically rotates 180° through the suction zone to the position shown in Figure 2 which *changes the direction of flow within the pump* without changing port positions. The idler carrier rotates in a cover casting fitted with stops so that the crescent can rotate only 180°—always through the suction zone. Upon resumption of counter-clockwise rotation, the crescent will swing back to the original position in Figure 1.

Tuthill Manufactures a Complete Line of Positive Displacement Rotary Pumps in Capacities From ⅓ to 200 GPM; for Pressures to 1500 PSI; speeds to 3600 RPM.

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NEW PRODUCTS

DRY FILM LUBRICANT IN KING-SIZE AEROSOL

A new, high-efficiency, non-inflammable colloidal graphite lubricant designated as a dry film material is being packaged in a giant economy-size industrial aerosol. Called "dgf 123," the colloidal graphite is carried in a 0.7% concentration at 34 to 38 psi (room temperature) in an instantly-drying Fluron propellant. Control elements carried with the synthetic colloidal graphite reportedly provide both an even dispersion during application and a high affinity for all clean, dry metal surfaces on contact, it is stated.

Uses of "dgf 123" are said to include application to prevent dry rubbing contact during initial movement of bearing surfaces and during subsequent run-in. Reported to lower interfacial tensions with oils, the dry graphite film causes frictional surfaces to retain thicker oil films. This wetting agent characteristic helps to distribute protective oil films to hard-to-reach areas and to hold them on the treated surfaces. Due to the fact that even dispersion of lubrication film is achieved with the aid of "dgf 123", smoother, more efficient bearing surfaces are maintained, the manufacturer states. Further information is available from Miracle Power Products Corp., 1101 Belt Line St., Cleveland 9, Ohio.

NEW PUMP MODEL MEETS ARMED FORCES STANDARDS

The Tuthill model CBGFK is one of a new series of pumps designed to meet Air Force-Navy Aeronautical design standards for use on ground equipment. The units are provided with a splined shaft, mounting pad, and porting to meet AN-D specifications. The new pumps are provided in two sizes, with capacities from 10 to 45 gpm at 150 psi and for speeds to 3600 rpm. They can be adapted to a wide variety of applications. The Tuthill Pump Co., 939 East 95th Street, Chicago, Ill., manufactures a complete line of positive rotary pumps.

HOSE ASSEMBLY MACHINE

A new power machine for speeding up assembly of reusable Hoze-lok fittings onto industrial rubber or cotton covered, wire or fabric braided hose is now available. Designated Model 422, the unit is portable, weighing approximately 160 pounds. It is driven by a ½ horsepower, 60 cycle electric motor. It readily handles hose ranging in size from ¼ to 1⅜ inches inside diameter. The machine is said to be simple to operate. The hose fitting socket is inserted in the chuck. The hose is clamped in the floating vise of the machine and pushed into the socket which is turned by the machine counter clockwise onto the hose. Then the fitting nipple is placed in the chuck and the socket (assembled onto the end of the hose) is clamped in the vise and pushed onto the nipple which then is screwed clockwise into the hose and socket. If full flow size hose is being used, a mandrel is screwed into the center of the chuck and the nipple slipped over the mandrel. The machine may be mounted on a bench, or lengths of pipe may be used to make legs for it. Complete operation and maintenance instructions are offered in Catalog 4707-A14, available from W. D. Wynant, Parker Fittings & Hose Div., Parker-Hannifin Corp., 17325 Euclid Ave., Cleveland 12, Ohio.

COUNTER IS CHECKING DEVICE FOR GREASING SYSTEMS

Panel mounted electric counters, fabricated by Durant Manufacturing Company of Milwaukee, Wisconsin, are installed with Trabon central lubricating systems as a method of checking the amount of grease being dispensed. The central lubricating system pumps grease to all parts of large machinery from one central point. The counter, mounted on the control panel, records each discharge of grease being fed to the machinery. The amount of grease being used, or the amount of grease remaining in supply, can be quickly determined by checking the counter. Through this method, if excess lubricant is being used, it may be spotted quickly and the master pump cycle control may be adjusted to compensate for the excess flow. A lubricating failure, such as decreased line pressure or power failure, may also be detected through regular checks of the counter. Complete information on the full line of Durant Counters is available through the Durant Manufacturing Company, Dept. LE, Milwaukee 1, Wisc.

BEAKERS WITHSTAND HI-TEMP

Beakers, molded from polypropylene for optimum high temperature service, have been announced. The beakers, because of the material used, reportedly have greater clarity and can thus be sterilized by autoclaving. They are available with capacities from 250 ml to and including 4000 ml. Agile-prolene beakers are chemically inert to practically all reagents at elevated temperatures and to most solvents at room temperature. Additional information may be obtained from the American Agile Corporation, P. O. Box 168, Bedford, Ohio.

The ladies of the Buffalo Section Cordially invite the wives of lubrication engineers to attend the

ASLE
Annual Meeting
Hotel Statler
Buffalo, N.Y.
April 21, 22, 23

(See page 82 for a complete Ladies Program)

ANNOUNCING...

the second BIG issue of

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Pacemaker T is a premium hydraulic oil with excellent heat resistant properties and suitable for every type of hydraulic system. These oils have high viscosity index to minimize viscosity change with changing temperatures.

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The Goetz Twins, owners of Stylette Plastics went into the plastics business in Pittsburgh in 1952 on the recommendation of their professor at Duquesne University. They started with $1,500 and bought a nine ounce injection molding machine to make plastic tile.

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QUALITY PETROLEUM PRODUCTS

ASLE MEMBERSHIP CLASSIFICATIONS

Membership in the American Society of Lubrication Engineers is in several grades as defined below. Assignment to grade is made by the Admissions Committee or Board of Directors on the basis of information submitted by the applicant.

Members: Members shall be persons not less than 24 years of age who: (1) are engaged in research and instruction at technical schools, universities, and various publicly and privately supported institutions in the field of lubrication; or (2) have occupied recognized positions as lubrication engineers for a period of three or more (not necessarily consecutive) years prior to date of admission, involving the responsibility for or supervision of the development, selection, field use and application of lubricants as differentiated from other activities; or (3) are indirectly concerned with the field of lubrication, but possessing other qualifications of experience, knowledge, and accomplishment, and have manifested a particular interest in the purposes and welfare of the Society to the extent that their membership would be a valuable contribution to the successful functioning of its activities. Fee 15.00.

Associate Members: Associate Members shall be persons less than 24 years of age, and those who do not completely fulfill the membership requirements for Members. Fee $7.50.

Sectional Sustaining Members: are such persons or organizations as may be interested in and desire to contribute to the support of the purposes and activities of a local Section of the Society. Fee $25.00.

Industrial Members: Industrial Members are such persons or organizations as may be interested in and desire to contribute to supporting the purposes and activities of the Society. Fee $150.00.

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NEW PRODUCTS

ROTARY HEAD MILLING ATTACHMENT

Precision engineered for use with horizontal or vertical milling machines, this new rotary head attachment reportedly has three exclusive features. These are (1) Direct Drive, featuring a silent V-belt drive direct to spindle with a choice of six speeds powered by a sturdy ½ hp, 1200 rpm motor (2) Double Rotary Worm Shaft, with front shaft for smooth, exacting rotation and a rear shaft for fast head rotation, and (3) Cross-Slide Screw which offsets the spindle from either end of slide. The manufacturer states that these three features are not available in any other comparable unit. The new Everede rotary head milling attachment has a large 10 in. pitch diameter worm gear and spring pusher on worm to assure constant engagement with no backlash, permitting heavy-duty climb milling cuts. Among other quality features of this unit are: Adjustable limit stops, built-in fast-action spindle lock, rotary degree scale, rotary dial graduated in one minute increments of a degree, cross-slide scale and a cross-slide dial. The Everede Rotary Head Milling Attachment rotates 360 degrees in either direction, and cross slide adjustment offsets spindle up to 3⅝ in. For further information and literature concerning the above, write, Everede Tool Company, 2000 N. Parkside Ave., Chicago 39, Ill.

PRECISION FORMING PROCESS

A precision forming process, the Minicast process, has been developed to combine the basic advantages of investment casting with those of such other precision methods as coining, cold forming, screw machine production and the like. The Minicast process is said to produce parts in almost all castable metals and alloys at production runs up to 50,000 pieces per day. In addition it provides dimensional tolerances as close as plus/minus 0.001 inch per inch; cross-sectional thickness as low as 0.020; superior surface finish (30-50 microinches, depending on the alloy); plus complex configurations. Additional information may be obtained from Casting Engineers, Inc., 2323 North Bosworth Ave., Chicago 14, Ill.

CONTROLLABLE CHECK VALVE

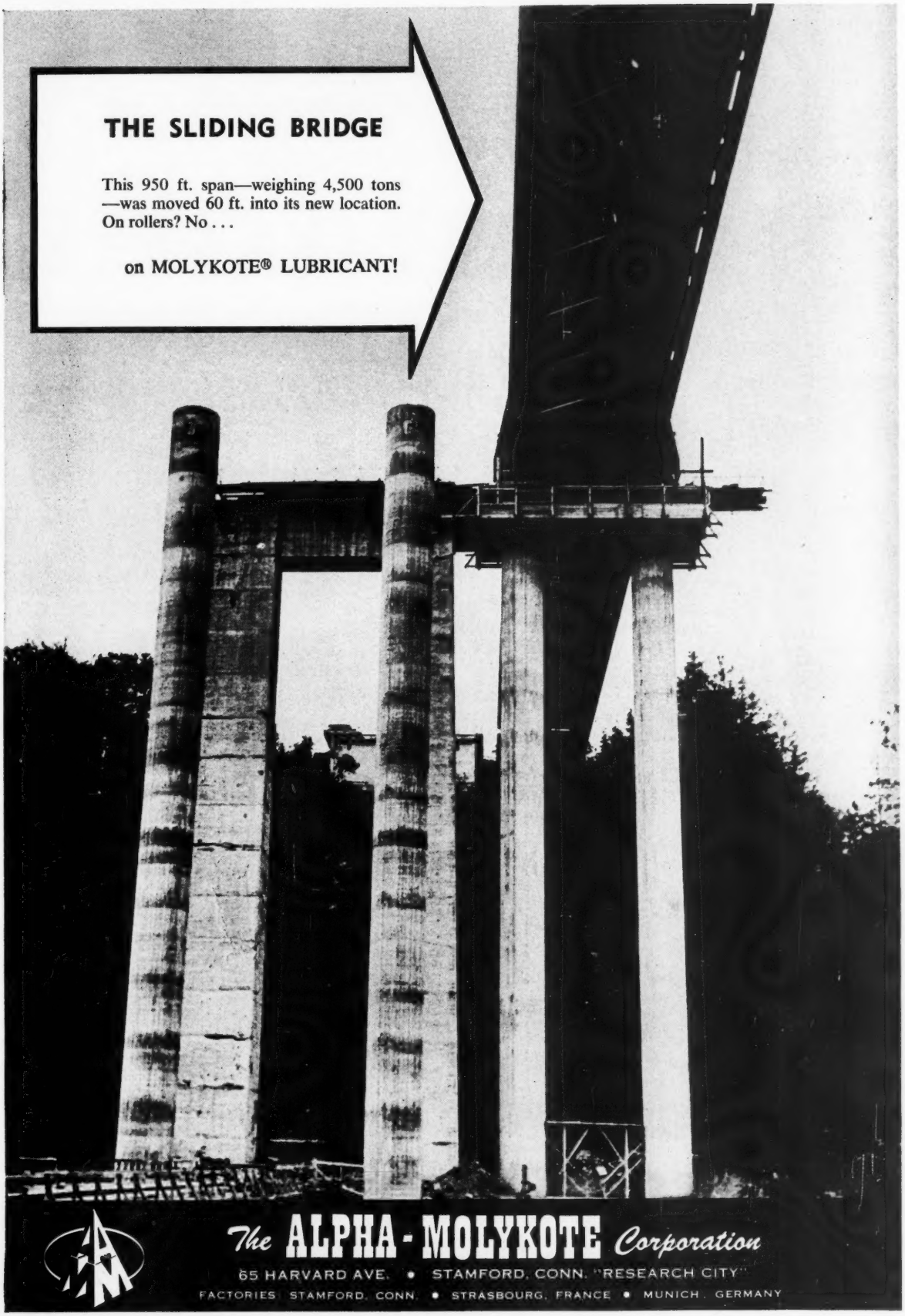
A new controllable check valve, which requires low mechanical force to unload is now available. Reportedly combining dependability and precision, the valve provides easy, positive control of systems in ground support equipment and many other exacting applications where safe and fast response is required. This spring loaded check valve permits free flow in one direction and checks reverse flow until it is unloaded by depressing the external plunger. The valve requires 100 lb. maximum pressure to unload with 3000 psi on the checked flow part. Additional features of this valve are its low pressure drop. With a 25 gpm flow at 3000 psi, the pressure drop is 65 psi maximum in the free flow direction and 35 psi maximum in the checked flow position. Zero external leakage and low internal leakage—measured in drops per hour with 3000 psi at checked port. Small in size, this valve weighs three and one-quarter lbs. Proof pressure is 4,500 psi; burst pressure 7,500 psi. While the valve is available with the above specifications, it can be modified to meet users specific requirements. Further information on this valve may be obtained from the manufacturer, Fluid Regulators Corp., 313 Gillette St., Painesville, Ohio.

NEW GREASE RESISTS LIQUID FUELS AND SOLVENTS

The lubrication of bearings and sliding surfaces in the presence of liquid fuels and solvents has long been a problem. Ordinary type greases are rapidly dissipated by the washing action of those liquids. Now, Pennsylvania Refining Company, Cleveland 4, Ohio, announces a new type grease highly resistant to most fuels and solvents. Its new solvent resistant grease is said to be impervious to the washing action of a wide range of petroleum, coal tar and chlorinated liquids. According to the manufacturer, the new grease permits normal lubrication of pumps, bearings and sliding surfaces that must work in the presence of solvents and liquid fuels.

SELF-LUBRICATING BEARINGS

Production of a new line of self-lubricating "Monoball" self-aligning plain bearings is announced by Southwest Products Co., 1705 Mountain Ave., Monrovia, Calif. Marketed under the registered trade name "Dyflon," the new bearings reportedly require no maintenance or service. The manufacturer recommends them for applications requiring low coefficiency of friction, long service life without lubrication and high dynamic load-carrying capacity.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Tentative Program for the Annual Short Course in Lubrication Engineering

Sponsored by the ASLE Education Committee
Chairman, M. E. Dougherty, Acheson Colloids Co.

Tuesday, April 21, 1959—LUBRICATION FUNDAMENTALS

9 a.m. to 12 noon

INTRODUCTION: Theodor Ranov, School of Engineering, University of Buffalo, Buffalo, N. Y.

The purpose of the course and a brief summary of the history and science of lubrication.

THICK FILM LUBRICATION: H. N. Kauffman, Research Engineer Bearing & Lubrication Section Westinghouse Research Lab. Pittsburgh, Pa.

Some of the principles of lubrication will be covered by lecture and demonstrations using models for illustration. Subjects treated include: Influence of friction under dry and lubrication conditions; development of the tapered oil film and its effect on friction in pivoted pad and journal bearings; oil film pressure and grooving; sparse lubrication, and principle of hydrostatic lubrication and its applications.

BOUNDARY LUBRICATION: E. M. Kipp—Director Research and Development—Foote Mineral Company, Berwyn, Pa.

When a fluid film of lubricant is unable to exist to keep two moving surfaces from contact, boundary lubrication must be depended upon to keep friction, wearing and scoring at a minimum. With no protective film, the surfaces will actually unite or bond together. To function as a boundary lubricant, a substance must reduce or prevent this metal-to-metal contact, by forming a film which will not flow out under the action of the contact pressure.

Tuesday, April 21, 1959—LUBRICANTS

2:00 P.M.-5:00 P.M.

PROPERTIES OF LUBRICANTS: Colonel Frank L. Gray, Gary Indiana Works, U. S. Steel Corp.

The properties of lubricants will be discussed and methods of measuring these properties will be described. Factors which influence lubricant performance will be treated including conditions that lead to failure of lubricated parts and analyses of such failures. The operation of lubricated parts will be related to lubrication theory so that practicing lubrication engineers will be able to utilize bearing theory to improve machine performance. Charles A. Bailey, National Tube Division, U. S. Steel Corp. Pittsburgh, Pa. will present a short discussion and distribution of U. S. Steel Corporation's "Lubrication Engineers Manual."

Application of Lubricants and Maintenance of Lubricating Equipment: Robert K. Ulrich, Steelton Works, Bethlehem Steel Corp.

A discussion of the various methods and equipment for handling and dispensing bulk lubricants, centralized systems, oil circulating systems and maintenance problems in connection with this type equipment.

SOLID LUBRICANTS: Arthur Stock, Manager Research & Development. Acheson Colloids Company, Port Huron, Michigan.

The use of solid lubricants in liquid media will be discussed. Properties of graphite, modybdenum disulfide and Tfe Resins will be pointed out and demonstrated by slides. Theory of thin film solid lubricants will be briefly talked about, and the use of dry film in practice will be illustrated.

FILM PRESENTATION: 7:30 P.M.-8:30 P.M. "BASIC PRINCIPLES OF LUBRICATION" Massachusetts Institute of Technology

Wednesday, April 22, 1959—GEARS—BEARINGS—SEALS & PACKINGS

9:00 A.M.-12:00 Noon

GEARS AND GEAR LUBRICATION: E. S. Reynolds, Senior Engineering Rep., Socony Mobil Oil Co., New York.

The principles of gear lubrication will be discussed. Also included will be examples of gear damage and how it is effected by lubricants and other operating or performance factors.

BEARING & BEARING PROBLEMS: Jack Gustafson, Martin-Rockwell Corp., Jamestown, N. Y.

Discussion will have emphasis on bearings, their design and some of the common causes of trouble.

BEARINGS—MAINTENANCE & LUBRICATION: Norman Bracht, Alan Wood Steel Co., Conshohocken, Pa.

(Cont. on p. 119)

Lube Lines

by A. F. Brewer*

THE EFFECT OF WEATHER

The weather is something like death and taxes—you just can't get away from it. A subject of conversation or a news item, yes, but nobody can do much about it. Just as weather affects our relative comfort, so can it also affect lubrication. Fortunately in the case of lubrication, something can be done about it if we plan accordingly, respect statistics and design for lubrication. Machinery which is operating outdoors on construction projects, in mining, on ore docks or in conveyor service is mostly affected. Complete protection of the functioning gears, bearings, power units and wire rope is impracticable because they are exposed to winds, driving rain and sleet or snow, and widely varying temperatures.

The designer plans accordingly, but he cannot fully anticipate the severity of the operating conditions. Materials handling machinery is tailor-made to some extent, but when it goes into service there is no telling whether it will wind up on a highway job in the southwest or in the iron ore range. The power shovel or dragline can become as nomadic as some contractors, going wherever the job is to be done.

THE MAINTENANCE PROBLEM

Maintenance can thus become quite an item in figuring the cost of operation in terms of, say, tons of material handled per day. If several days of idleness are necessary in order to obtain and renew a vital part, the monetary loss could become alarmingly high—often running into thousands of dollars. Maintenance also is something like the weather, it will always be with us. There is no final solution for this problem but there is a means of alleviation—lubrication.

Far-sighted management takes all this into consideration, whether the machinery is a fixed unit such as a lengthy belt conveyor in the ore fields or a self-propelled power shovel in service on a super highway. But does management always realize how vital that seemingly insignificant item in overall expense—the cost of lubrication—can become? Take cold weather service for example, especially where the machinery must be started and stopped daily. Freezing may occur if the mercury takes a sudden drop when a cold front develops. We must remember that freezing can involve more than ice and snow. Lubricants also can freeze when subjected to low temperatures. Under extremely low temperatures certain metals also can become brittle and subject to fracture.

WHAT'S THE ANSWER?

What can the operator do about this? Well, let us assume that he knows that he has a machine which has been scientifically designed, with adequate factors of safety for any emergencies. Let us also assume that he knows that the builder has studied the adaptability of the various means of lubrication available and that the systems on his machine are capable of furnishing adequate lubrication. What next? Why, the lubricants, of course.

In cooperation with machinery builders and designers of modern methods of lubrication, the chemist and lubricant manaufacturer have become active partners in the project of producing lubricants capable of meeting extreme service conditions. There was a time when low pour point in an oil and penetration in a grease were the primary criteria in refining for low temperature service, just as high flash point was considered adequate for high temperatures. Today all this has changed. Heavy-duty motor oils, fluid wire rope lubricants and all-purpose greases are significant examples. These are the tools which weather-conscious management must fit into the pattern of service conditions. Some may say "They cost more!" So what? They cost more to produce because:

1. They are the result of exhaustive laboratory research in the study of additives and compatability.
2. They have been perfected after months of field testing under the same conditions of weather and temperature as may confront the user.
3. They are scientific blends of the most suitable components.

But let's face it. Do they cost more in proportion to the cost of the machinery they are designed to protect? Do they cost more when this cost is broken down in terms of the cost or value of the materials handled? When we consider that dependability is built into the modern lubricant equally as scientifically as it is built into the modern machine, we realize that today it is a question of buying lubrication, not just lubricants.

Management cannot control the weather but it can exert a definite influence over weather's effect on machine performance and dependability by using as good a judgment in choosing lubricants as in buying machinery.

* Consultant and author of "Basic Lubrication Practices."

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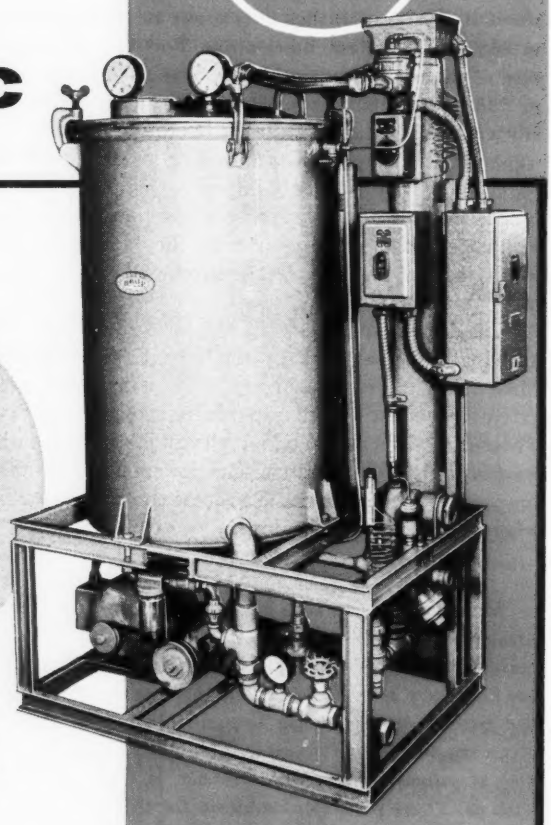

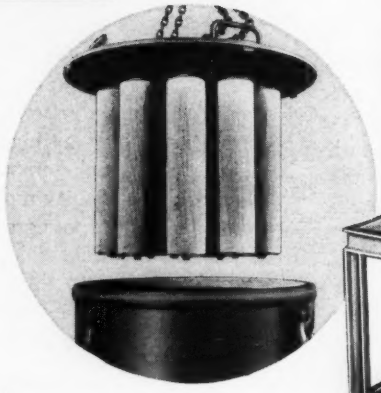

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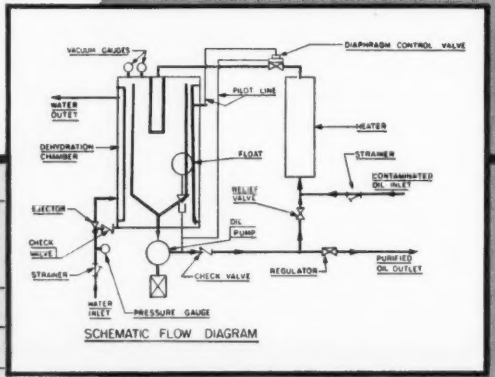

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Vol. 15, No. 3

Lubrication Engineering

a door to knowledge

Back around the time we were in grammar school, a tremendous value was placed on a number called our I.Q.—our intelligence quotient. We stood a little in awe of this number because we believed that it illustrated the sum and substance, not only of our learning, but of our native ability.

Time has proved that that little number really does not have all the meaning that our teachers ascribed to it. For it could not possibly take into account all the variables that go into the shaping of a man. We have since discovered that a man's learning cannot be reduced to a single number or even to an equation. For a man's knowledge increases daily, it is a variable dependent upon his ability and desire to learn as well as the opportunity available for learning.

Learning does not stop with the presentation of a high school or a college diploma. It is a continuous process which lasts for as long as a man desires to have it last. And sometimes it is thrust upon him unwillingly when he least desires enlightenment.

In our age, the mass media make knowledge available to all. But it is a general kind of knowledge. To get the specialized knowledge that many of us desire, it is necessary to look farther than the daily newspaper, the radio, the television, or the consumer magazine. Those of us in the specialized field of lubrication engineering have an excellent source of information—this Society exists to disseminate knowledge in our field. The most excellent opportunity for availing ourselves of this knowledge comes yearly in April when ASLE holds its annual meeting. This year, as in previous years, the papers to be presented are of so varied a nature as to provide information for all those who hear them, no matter how much knowledge of the field they may already possess.

We cannot urge you too strongly to take advantage of this opportunity to swell your own fund of learning, to accumulate facts from the storehouse of information which will be thrown open to you at this meeting.

J. D. Lykins, President

Investigation of Factors Affecting High-Temperature Gear Operation

by E. E. Shipley
General Electric Co., Lynn, Mass.

For the past several years, interest has been keen in regard to high temperature gear operation. With the advent of supersonic planes and missiles, more and more effort has been directed toward solving the engineering problems associated with this new environment.

A full scale high temperature gear testing facility was put into operation in August 1956. Since then, a considerable amount of testing work has been done to find lubricants, materials and gear designs that will operate successfully at high temperatures. This report covers some of the findings.

The aircraft industry, as a whole, was one of the first groups to become deeply involved in high temperature gear operation. On the surface, it seemed that new and improved lubricants might be the answer, but if temperatures increased much beyond 350F, the gear material would require investigation.

From an engineering point of view it was agreed that the only way to find the correct solution to the high temperature problem would be to build a full size, full load gear test facility and duplicate expected field conditions on a test stand. In this manner, information could be gathered on lubricants and materials as well as over-all system performance.

The test setup is essentially a four square gear torque testing device whereby one gear box is loaded against another similar gear box, Fig. 1. A driving motor furnishes the static and dynamic friction losses of the two rotating gear boxes. The motor, the speed increasing gear box and the loading gear box are of conventional design and operate at approximately room temperature. The test gear box or "hot box," is of dual construction and can be heated to 700F and higher. The inner casing contains the test gear, pinion and inlet oil nozzle. High temperature seals prevent leakage to the outer casing. The outer casing supports the test gear and pinion shafting and includes a bearing cooling system to keep bearing temperatures at a safe operating level. The heating elements are located between the inner and outer casing.

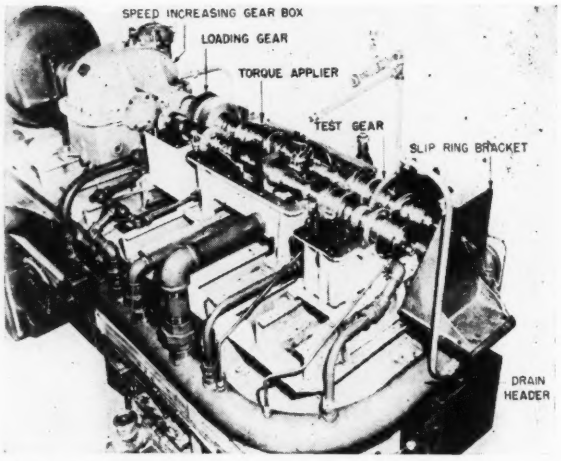

Fig. 1. High temperature gear test stand.

The design of the test stand was based upon simulating typical aircraft gearing practice with respect to center distance, pitch, speed, ratios and gear loading.

TABLE 1. Capacities of the High Temperature Gear Test Machine

Maximum Pinion Speed	24,000 rpm
Maximum Tooth Loading	* 2,000 *K*-factor
Maximum Simulated Power	1,700 hp
Center Distance	5.000 inch
Diametral Pitch (Test Gears)	5 to 12
Ratio	1:1 to 3:1
Maximum Test Box Temperature	700F plus
Maximum Contacting Face Width	3/4 inch
Load Variation	Variable at any speed

$$*K\text{-factor} = \frac{W}{F_e \times d_p} \times \left(\frac{m_G + 1}{m_G}\right)$$

where: W_t = tangential driving force, pounds
F_e = contacting face width, inches
d_p = pitch diameter of pinion
m_G = gear ratio (m_G = 1 or larger than 1)

Presented at the Annual Meeting of the American Society of Lubrication Engineers held in Cleveland, Ohio, April, 1958.

In addition to designing standard test gear specimens, the machine was made versatile enough to permit many variations of gear designs to be tested providing the center distance remained constant. The capacities of the high temperature, full load gear test machine are given in more detail in Table 1. Once the full scale, full load gear test facility was in operation, available testing time was at a premium. A considerable amount of laboratory material and lubrication evaluation had be be undertaken to screen out promising candidates before they reached the high temperature gear testing phase to conserve the available test time as much as possible.

The lubricant phase consisted of running bench tests at elevated temperatures on:

1. Oxidation stability
2. Thermal stability
3. Volatility
4. Lubricity

More than 40 possible high temperature lubricants have been evaluated on the screening bench test facilities. These fluids are products of 20 different manufacturing concerns and represent a wide variety of chemical compounds.

The oxidation stability test measures the stability of the oils at elevated temperature in the presence of metal catalysts and air towards the formation of acidic compounds, polymerization with resultant viscosity increase or deposition of sludge and corrosion of the metals present. The oxidation tests were run by bubbling dry air through thirty grams of test fluid at the rate of three liters per hour for six hours. A high air to oil ratio was maintained which is indicative of aircraft gearing practice. The metal catalysts, which were placed in the apparatus, were pieces of nitrided steel, tool steel, and carburized steel. (These steels are potential high temperature gear materials.)

The results of a number of oxidation tests performed at 500F are given in Table 2. All of the fluids in Table 2 exhibited better oxidation stability than MIL-L 7808C diester. It is evident that the silicone group and the chlorinated biphenyls were outstanding when compared with MIL-L 7808C diester.

Additional oxidation test results are tabulated in Table 3. These fluids exhibited oxidation stability that was inferior to MIL-L 7808C. Note—there are several grades of diesters and mineral oils that have poor oxidation stability.

Thermal stability tests measure the ability of the fluids to withstand high temperature in the absence of oxidizing influences. These tests were run using similar apparatus and procedures as used in the oxidation tests except that nitrogen was substituted for air. The results obtained point out the advantage that would be gained by excluding air from the lubricating systems of aircraft gear boxes. In general, an improvement was indicated in most every case; the large increase in viscosity, neutralization number and sludge formations associated with the oxidation tests were not present in the thermal stability tests. Some of the fluids tested on the thermal stability tester are presented in Table 4. The silicones and the chlorinated biphenyls still look good. The inert atmosphere greatly improved the per-

TABLE 2. Fluids that Exhibited Better Oxidation Stability than MIL-L 7808C at 500°F
Catalyst Metals – (1) Nitrided Steel, (2) Tool Steel, (3) Carburized Steel Test Time 6 Hours

Test Fluid	Liquid Loss %	Viscosity Change %	Neutralization No Before	Neutralization No After	Sludge Formation %	Gain in Wt. Catalyst Metals, mg: Metal 1	Metal 2	Metal 3	Catalyst Condition
Methylphenyl Silicone	Nil	4	Neut.	Neut.	Nil	.4	.2	.2	Very slight stain
Chlorinated Biphenyl	1	2	Neut.	Neut.	Nil	.8	.7	.7	Yellow stain rust
Methylphenyl Silicone + Add.	.2	1	Neut.	Neut.	Trace	.1	−.1	−.1	Blue amber stain
Chlorophenyl Silicone	3	11	Neut.	Neut.	Nil	1.1	1.1	.6	Very slight stain
Fluoroester	2	−17	Neut.	4.8	Nil	.5	.7	.8	Blue stain
Silicate Ester	4	−20	.02	4.3	Nil	.2	.1	.2	Very slight stain
Diester # 1	1	34	Neut.	38	2.5	.2	.2	.1	Slight stain
Polyether	2	4	.50	8.4	6.9	.5	.8	.5	Black
Polybutene	1	19	.28	3.4	5.9	5.5	2.3	4.4	Heavy tacky coat
Mineral Oil #1	3	1	.30	1.0	8.9	.6	.5	.4	Stain
Diester #2	4	10	.06	38	3	.2	.4	.3	Stain
Pentaerythritol Ester	5	38	.03	51	Trace	.6	0	.2	Slight stain
Diester MIL-L 7808C	4	24	.21	29	9	.5	.5	.5	Black stain

formance of Mineral Oil #2 and MIL-L 7808C by reducing the amount of sludge formation and change in neutralization number.

The fluids that seem to have the best oxidation stability at high temperatures, (500F) are the silicones and chlorinated biphenyls. The fluids that seem to have good thermal stability are the silicones, chlorinated biphenyls and mineral oils. The mineral oils and chlorinated biphenyls have high volatility and poor viscosity temperature characteristics. (They also have poor low temperature properties.) Therefore, the silicones undoubtedly stand the best chance of surviving at elevated temperatures. The main drawback with the silicones is their lack of good lubricity when checked with the many conventional bench wear testers.

Considerable efforts have been made in the direction of producing a lubricating surface on materials to be used in conjunction with silicone oils. A number of different treatments (Table 5) have been evalu-

TABLE 3. Fluids that Exhibited Inferior Oxidation Stability than MIL-L 7808C at 500F
Catalyst Metals – (1) Nitrided Steel, (2) Tool Steel, (3) Carburized Steel Test Time 6 Hours

Test Fluid	Liquid Loss %	Viscosity Change %	Neutralization No Before	Neutralization No After	Sludge Formation %	Gain in Wt. Catalyst Metals, mg: Metal 1	Metal 2	Metal 3	Catalyst Condition
Diester MIL-L 7708C	4	24	.21	29	9	.5	.5	.5	Black stain
Mineral Oil #2	5	96	.12	9.2	5.3	−.9	−4.7	−2.1	Black pitted
Diester MIL-L 25336	1	30	.19	26	7.6	+.3	.3	.3	Stain
Mineral Oil #3	1	145	.11	1.0	7.4	−.1	−.9	.3	Stain
Silicate Ester #2	6	−165	Neut.	16.3	3.0	1.3	.5	.2	Black stain
P, P′-Di (M-Cresyloxy)	2	1270	.45	16	2.1	−.9	.5	.2	Stain
Diester #4	1.5	60	.03	26	5.5	−1.6	−.6	−.2	Stain small pits
Neopentyl Glycol Dibenzoate	4	15	.17	54	1.4	−17.4	−3.1	−10.0	Stain pitting
Chlorinated Hydrocarbon	19	70	Neut.	6	3.8	2.7	−1.2	.2	Stain rusting
Mineral Oil #4	7	272	.06	6	5.6	−2.3	−4.7	.4	Stain
Methylated Wax Phenol	7	59	.60	14	20	−5.0	−24.0	−8.0	Pitting rusting
Phosphate Ester #1	Nil	390	.06	75	26	3.8	1.9	1.2	Heavy black deposit
Phosphate Ester #2	2	394	.02	84	24	4.0	4.1	1.7	Heavy black deposit

TABLE 4. Thermal Stability of Fluids at 500°F Catalyst Metals – (1) Nitrided Steel, (2) Tool Steel, (3) Carburized Steel (Entire System Blanketed with Nitrogen) Test Time 6 Hours

Test Fluid	Liquid Loss %	Viscosity Change %	Neutralization No Before	Neutralization No After	Sludge Formation %	Gain in Wt. Catalyst Metals, mg: Metal 1	Metal 2	Metal 3	Catalyst Condition
Methylphenyl Silicone	Nil	1	Neut.	Neut.	Trace	.1	.4	0	Slight amber stain
Chlorinated Biphenyl	1	5	Neut.	Neut.	Nil	.4	.4	.4	Slight amber stain
Mineral Oil #2	1	6	.12	.20	Trace	.3	.2	.2	Slight amber stain
Chlorophenyl Silicone	Nil	8	.10	.10	.3	.3	.5	.4	Slight amber stain
Diester MIL-L 7808C	1	26	.16	1.4	.2	.4	.1	0	Slight amber stain
Diester #2	2	6	.06	1.7	Trace	.4	.3	0	Slight amber stain
Pentaerythritol Ester	1	.5	.03	1.1	.05	.2	.4	.7	Slight amber stain
Silicate Ester #1	Foam	1350	.20	.56	.23	.6	.7	.5	Slight amber stain

TABLE 5. FALEX TESTS. SURFACE TREATMENT RESULTS WITH SILICONE OIL (BOTH THE TEST PIN AND V BLOCK WERE COATED)

SURFACE	ULTIMATE SEIZURE TEST LOAD LB
Steel	450
Phosphate	600
Chromized	600
Platinum	600
Molybdenum Disulfide	800
Nickel	1100
Chromium	1500
Silver Chloride	1500
Cobaltous Chloride	1500
Ferrous Chloride	1500
Copper	2400
Cadmium	2500
Sulfurized	2500
Gold	3000
Zinc	3250
Indium	3400
Silver	4500

ated on the Falex tester in the presence of gear materials and silicone lubricants. Of these various treatments, silver plate appears to be promising. Results have shown that silicone oil will provide satisfactory lubrication for numerous metal combinations. Steel on steel gives the worst results. Break this combination and improved lubrication will result. One basic limitation to the use of surface coatings is the fact that if they are worn away during sliding, they usually cannot be replaced.

Once operating temperatures exceed 350F, the conventional case carburized gear materials are no longer adequate for the job at hand. Materials that will operate at the elevated temperatures and still retain most of their room temperature properties are required. The criteria for a good high temperature gear material are:

1. High fatigue strength at elevated temperatures (as well as at normal operating temperatures).
2. High hardness at elevated temperatures.
3. Good resistance to scoring and pressure welding.
4. Good machinability.
5. Absence of embrittlement from aging at temperature.

With these properties in mind, a large group of possible high temperature gear materials were screened by running high temperature tests on the pneumatic fatigue tester, the hot hardness tester, the Charpy impact tester and the pinion fatigue tester. The chemistry of many of these materials is coded and listed in Table 6.

Table 7 shows some results of the pneumatic bar fatigue tester. This test uses a notched specimen 2¼ in. long by approximately .40 in. square. The tests are performed under fully reversed beam bending at 600F. Many varieties of steel were tested. However, this table gives a sample of the highest results obtained.

Non-rotating pinion fatigue test specimens were made from some of the interesting materials indicated by the results of the pneumatic fatigue tester. These pinion specimens were tested on a Sonntag universal fatigue machine modified for 600F operation. This type of test is not a running test but a tooth breakage test. A given tooth is subjected to unidirectional loading by meshing a tooth against a loading device and exerting a torque on the pinion by means of a lever coupled to the fatigue machine's mechanical oscillator. The pinion simulates to some extent the tooth proportions of a typical aircraft pinion and can be seen schematically by referring to Figure 2. The tooth cutting data for the test pinion can be found in Table 8.

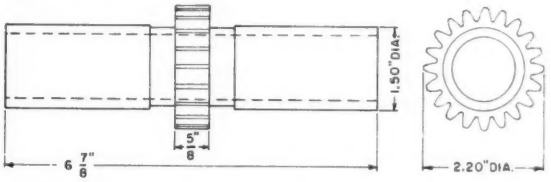

Fig. 2. Tooth breakage fatigue test pinion.

TABLE 6. CHEMISTRY OF MATERIALS

MATERIAL CODE		C	M_n	P	S	S_i	C_r	N_i	M_o	W	V	OTHER
1	A	.21	.63	.012	.012	—	1.00	3.26	.26	—	—	1.31 AL
2	J	.84	.26	.028	.019	.34	3.72	.14	4.86	5.92	1.92	
3	D	.099	.45	.016	.022	.31	12.64	.26	.06	—	—	
4	F	.12	.70	.021	.016	.44	18.40	9.20	.28	—	—	
5	K	.28	—	—	—	—	3.25	—	—	9.00	.25	
6	L	.19	—	—	—	—	1.50	3.00	5.00	—	—	
7	M	.40	.71	.012	.006	1.64	.87	1.81	.38	—	.08	.027 AL
8	S	.35	.30	—	—	1.00	5.00	—	1.16	1.30	3.00	
9	R	.55	.30	—	—	.95	4.95	—	1.2	1.20	—	
10	T	.24	.24	—	—	.15	.37	5.0	.25	—	.10	1.75 AL
11	U	.40	—	—	—	1.05	5.00	—	1.35	—	.35	

Pinion fatigue tests were conducted at room temperature, 300F and 600F. Some of the fatigue results of possible high temperature materials that have been heat treated in various ways are presented in Table 9. Table 9 gives the apparent endurance limit for the pinion teeth in terms of unit load.

Unit load is a common expression used in the gear trade and can be calculated for the pinion test specimens as follows:

$$W_u = \frac{W_t}{F_e} \times P_d$$

where: W_u = unit load, psi.
W_t = tangential driving force, pounds.
F_e = effective face width, inches.
P_d = diametral pitch.
since: for the test specimens
$P_d = 10$
$F_e = 5/8$
then: $W_u = 16 \times W_t$

Once the endurance limit for a given material has been determined in terms of unit load, then this information can be used for any standard gear design by solving the equation for unit load.

It is quite interesting to note from Table 9 that some of the materials indicated approximately the same unit load endurance limits at 600F as they did at room temperature. Induction-hardened 2J material indicated a decrease in unit load at room temperature as compared to the 600F values. Some of the materials that produced the highest values of mean endurance limit were also quite good at room temperature. In general, the results show that high temperature materials can be found and heat treated in a satisfactory manner. Undoubtedly, there are other materials and heat treatments which have better fatigue and wear characteristics than these materials and treatments evaluated by the screening tests. Therefore, it is quite important that investigations continue along this line.

Hot hardness tests were made on all of the various material samples at 400F, 600F and 700F. The high temperatures reduced the hardness levels from that at room temperature. However, most of the material samples had ample hardness at the elevated temperatures.

One property of the various materials that seems somewhat disturbing is the impact strength. V-notch Charpy impact tests have been run at room temperature and 500F. Most of the material candidates for high temperature work gave low foot-pound impact readings at room temperature. At elevated temperatures, impact readings improved. Some of the better fatigue resistant materials at high temperature appear to show little improvement over the low room temperature values, however.

The interest of the author's company in lubrication and materials ultimately is in the effect they have at elevated temperatures on the long range wear and scoring type of gear tooth failures. In addition to this, of course, the over-all operation of the high temperature test stand is very important. It had been predicted that, at elevated temperatures, many of the fluids would exhibit a drastic tendency to sludge. Most of the oils tested did form a tremendous amount of sludge and gummed up the whole lubrication system. Sludge and varnishes were deposited on the inside of the hot

TABLE 7. Pneumatic Bar Fatigue Tests
Notched Specimens at 600F

MATERIAL (CODE TABLE #6)		HEAT TREATMENT	APPARENT ENDURANCE LIMIT*
1	A	Nitrided	155
2	J	Thru-Hardened	120
2	J	Nitrided Thru-Hardened	160
3	D	Carburized	95
3	D	Nitrided	80
4	F	Nitrided	75
5	K	Carburized	125
6	L	Carburized	90
7	M	Thru-Hardened	90
8	S	Carburized	130
9	R	Thru-Hardened	100
8	S	Nitrided	175
10	T	Nitrided	170
11	U	Nitrided Thru-Hardened	155
11	U	Nitrided Med. Hardened	120

*Apparent endurance limits in 1000 psi; average of 2 or 3 specimens tested; assuming full stress concentration of 1.0 applies.

TABLE 8. Cutting Data Pinion Fatigue Tests

Number of Teeth	20
Diametral Pitch	10
Pitch Diameter	2.000
Addendum	.100
Whole Depth	.240
Circular Pitch	.31416
Tooth Thickness—Chordal	.154/.152
Root Diameter	1.740
Helix Angle	0°
Pressure Angle	20°
Material	Spec. by Engineer
Heat Treatment	Spec. by Engineer

TABLE 9. High Temperature Pinion Tests
Apparent Endurance Limit (Unit Load)

MATERIAL (CODE TABLE #6)		HEAT TREATMENT	UNIT LOAD ROOM TEMP	UNIT LOAD 300F	UNIT LOAD PSI 600F
2	J	Thru-Hardened	—	—	23
2	J	Thru-Hardened (Finish Ground)	—	—	30
2	J	Induction Hardened	33	32	44
2	J	Induction Hardened Nitrided	—	—	44
1	A	Nitrided	46	46	43
1	A	Nitrided (Heavy Case)	—	—	40
11	U	Nitrided (Med. Hard Core)	43	—	44
8	S	Nitrided (Fully Hard Core)	50	—	53
10	T	Nitrided	55	—	54

gear box as well as on the test gears themselves. The filter element at times would clog up and completely restrict the oil flow. Since engine designers are quite interested in not making drastic design changes in their basic conventional lube systems, it was decided that in spite of the obvious difficulties, filters would be used in the lube system.

Twenty-five hour endurance tests were run on the high temperature gear endurance test stand. The inlet lubricant temperatures were maintained at 400F and the gear blank temperatures were maintained at 500F. The load on the test gears was set at 700 K-factor and the tests were run at a speed of 10,000 ft. per minute pitch line velocity. The oil flow to the gear mesh was approximately 2/10 gpm. The results of a series of these gear tests are presented in Table 10. From an operational point of view, the silicones, with the exception of the methylphenyl silicone, performed much better than most of the other types of fluids. The silicones, however, revealed evidence of wear and perhaps in the long run, this may cause considerable concern.

Fig. 3 shows some comparative wear tests made with nitralloy test gears and several types of synthetic lubricants. These test results were obtained by using a radioactive technique of detecting the amount of wear. The silicone, (a chlorophenyl) and the high temperature diester were run with 400F inlet oil temperature and 500F gear blank temperature. For comparison purposes, a MIL-L 7808C diester was run at 300F inlet oil temperature and 310F gear blank temperature. The MIL-L 7808C diester at 300F gave excellent performance from a wear point of view. The silicone oil test indicates on a comparative basis that it initially experiences approximately eight times as much wear as

TABLE 10. High Temperature Gear Endurance Test

LUBRICANT	GEAR MATERIAL	LUBE TEMPERATURE	GEAR TEMPERATURE	RESULTS
1. Diester (MIL-L 7808C	Nitrided 1 A	300F	310F	Sludged slightly No gear wear
2. Diester #2	"	400F	500F	Sludged badly Test stopped after 7 hours
3. Silicon (Chlorophenol)	" "	400F	500F	No sludge Some gear wear
4. Silicone + Add. (Methol-phenol + Add.)	"	400F	500F	Sludged mod. Slight gear wear
5. Silicone (Metholphenol)	"	400F	500F	Gear scored badly before 7 hours
6. Silicone (Methol + Metholphenol)	"	400F	500F	No sludge Some gear wear
7. Mineral Oil #3	"	400F	500F	Sludged badly Slight gear wear

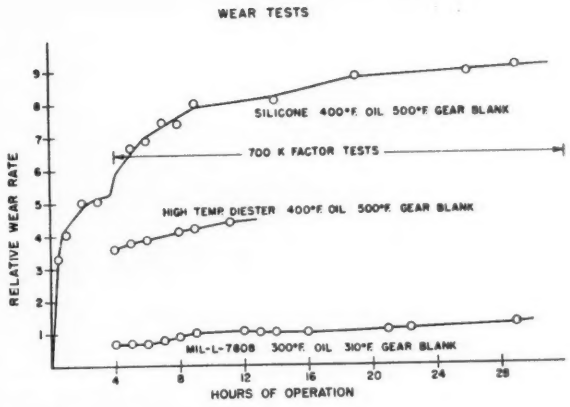
WEAR TESTS
RELATIVE WEAR RATE
SILICONE 400°F. OIL 500°F. GEAR BLANK
700 K FACTOR TESTS
HIGH TEMP DIESTER 400°F. OIL 500°F. GEAR BLANK
MIL-L-7808 300°F. OIL 310°F. GEAR BLANK
HOURS OF OPERATION

Fig. 3. Radioactive comparative wear tests.

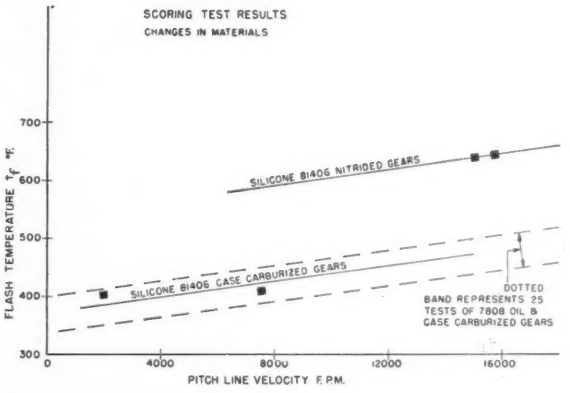
SCORING TEST RESULTS
CHANGES IN MATERIALS
FLASH TEMPERATURE °F.
SILICONE 81406 NITRIDED GEARS
SILICONE 81406 CASE CARBURIZED GEARS
DOTTED BAND REPRESENTS 25 TESTS OF 7808 OIL & CASE CARBURIZED GEARS
PITCH LINE VELOCITY F.P.M.

Fig. 4. Material comparative scoring tests.

the MIL-L 7808C. After a period of time, the silicone oil wear rate levels out to approximately 2 mg/hr. The MIL-L 7808C has very little wear initially and after a period of time, it settles down to a wear rate of 1 mg/hr. or approximately ½ the rate of the silicone oil.

The high temperature diester test at the same operating temperatures as the silicone oil test seems to perform better than the silicone but not as good as the MIL-L 7808C lubricant at 300F. It would seem from the information gathered to date that the silicones have initially a high rate of wear but after the run-in period, they level out. Perhaps further tests may prove that silicone can perform satisfactorily for most design applications.

From a scoring point of view, the performance of many of these lubricants and metal combinations show promise. It is hard to determine which deserves the most credit, the material or the lubricant. It is quite obvious however, from Fig. 4 that a change from a case carburized gear material to a nitrided gear material produced a very large increase in the scoring resistance of the material-lubricant combination.

The dotted line in Fig. 4 represents the results of approximately 25 scoring tests of MIL-L 7808C oil

(*Cont. on p. 119*)

The War on Wear

by A. F. Brewer,
Consultant

Ever since time immemorial, it has been accepted that wear results from motion. Equally as acceptable has been the fact that lubrication is the most effective way to control the rate of wear. The ancients appreciated this when they resorted to animal fats to stop the squeak of chariot wheels; the age of steam introduced the tallow pot, the mechanics of lubricant application closely paralleled the development of the machine era. Today effective lubrication is the controlling factor in twentieth century mass production.

Deterioration, of course, accompanies wear. The most classic example is the erosion of rock by water. In the operation of industrial power plant or production machinery, wear can have a definite relationship to cost. As Melvin Mandell writing in *Dun's Review and Modern Industry* has so aptly stated, "Wear is a chain reaction" which can lead to "replacement of costly parts, shortened equipment life, and profit-eating down-time on some key production machine or a disastrous wreck" on transportation equipment.

MANAGEMENT INTEREST

Management in modern industry is taking a new look at the staggering cost of wear. Mandell calls it "Industry's War on Wear," the balancing of the cost of wear against the cost of lubrication. The idea of waiting until something happens and then doing something about it is archaic. Management today locks the barn door before the horse is stolen. In other words, the war on wear is being entered into at the specification stage, thus making sure that the best anti-wear devices have been designed into their equipment. As Mandell points out, the temptation to ease the budget problem by skimping on anti-wear devices in equipment specifications in this age of speedup mass production can too often turn out to be false economy.

The appreciation of the cost of wear and the respect for the vital part which lubrication plays as a safety factor is creating a virtual corps of lubrication specialists among top management people. Since these men have the responsibility for choosing lubrication systems best suited to their service conditions and justifying the initial cost of such systems, they must be prepared to back up their judgement with figures relating to the cost of maintenance. The logical step is a consultation with the plant lubrication engineer. If none is available, one should be engaged if annual maintenance costs exceed $100,000. Thus management can speak with authority on matters such as the economics of centralized lubrication, the protective value of detergent and other additives, the control of fire hazards in hydraulic systems, the location of lubricating devices, and various other factors which need to be understood thoroughly if the finally-accepted systems and lubricants recommended are to establish the desired line of resistance in this war on wear.

INFLUENCE OF OPERATING CONDITIONS

But effective lubrication can go only so far, it is not the final answer. Mandell suggests that it constitutes about three-fourths of the whole anti-wear story. The rest of the headaches involve extremes of operation such as overload and wide variations in temperature, weather conditions, exposure, misuse, neglect or faulty design. Under such conditions, lubrication cannot be a panacea. Heat can exert a destructive influence on a wearing surface and lubricant. Sometimes the actual structure of the metal can be changed, sometimes the resultant accelerated oxidation rate may cause the bearing metal to break down, or it may materially reduce the lubricating ability of the lubricant. Overload, in turn, can cause deformation of certain types of bearings, or it can seriously change gear tooth conformation.

The war on wear can be carried on more successfully when the management—lubrication—specialist has studied every possible operating condition which could later influence effective lubrication. Here it is well for him to get down to ground-level and go out into his own plants, or others, to evaluate these conditions based on the experience of men who have to live with them, i.e. the machine operators, the maintenance men, the machinists, the millwrights. They all realize that, when a machine is operating dependably and is adequately protected by lubrication, their jobs are more pleasant. When management gives them the means to keep things running most efficiently, they will have more time to become part of the team and come up with suggestions for even greater reliability.

As Mr. Mandell concludes, "The war on wear will never end, because as long as any machine is doing useful work—producing or hauling the products of industry—it is wearing out. Equipment designers, researchers, and maintenance men are in the front lines of this fight, but they can't hope to win without proper management leadership and support."

GUARDING YOUR EQUIPMENT INVESTMENT— A MANAGEMENT CHECKLIST

☐ Does one man have responsibility for lubrication and anti-wear measures in equipment and/or vehicles? Is he a lubrication specialist? (A good rule of thumb: hire one if maintenance costs are over $100,000 annually.)

☐ Are careful records kept to insure compliance with lubrication specs and schedules?

☐ Is there a maintenance card, including lubrication requirements, on each machine or vehicle?

☐ Do your maintenance people keep good records on replacement of wearing parts? (You may have an excessive wear problem and not know it.)

☐ Is your plant subject to considerable machine downtime because of manual lubrication methods? (Many types of machines must be stopped for manual lubrication under stringent safety rules.)

☐ Have scrap rates increased lately? (Too much wear lowers machine accuracy; poor lubrication makes machines run erratically.)

☐ Must machines be disassembled for lubrication? (Modern practice is to eliminate inaccessible wearing points.)

☐ Do your machinery purchasers insist on central lubricating systems, sealed or self-lubricating bearings and wear-resistant parts for new machines, where practicable?

☐ Are there safety interlocks between the central system and all machines serviced by it?

☐ Do your maintenance men avail themselves of the field services of machinery, bearing, and lubricant suppliers? (Most of it's free and usually not linked to a sales pitch.)

☐ Are manufacturer's specs on lubricants for every new machine or vehicle checked against current stocks of lubricants for equivalents?

☐ Do your operating personnel buy lubricants by price only? (A very dangerous practice, considering the low cost of lubricants as compared to the total cost of machines.)

☐ Are lubricant stocks checked at regular intervals for simplification? (Annual checking is essential, and semi-annual is recommended.)

☐ Are lubricants exposed to air? (Wherever possible, lubricants should be transferred directly from shipping container to wear point to avoid dirt contamination.)

☐ Are bearings over-lubricated? (It takes amazingly little oil to lubricate a bearing. Most of the damage to small electric motors in industry is done by over-lubrication.)

☐ Do your maintenance men routinely change the type of lubricant when a bearing fails, or do they make a thorough investigation of the failure? (There are dozens of reasons for bearing failure, the most prominent being overloading and dirt.)

☐ Are lubricants stored indoors where there is less chance of contamination? Is the storage room kept clean, suitably drained, and made of fire-resistant materials?

☐ Are the older lubricants used first?

☐ Are lubrication men encouraged to report potential trouble spots to the maintenance force?

☐ Are grease-gun fittings standardized, or is a multiplicity of grease guns required?

☐ Are lubrication lines well secured and located so as to discourage their use as ladders or stepping points for operating or maintenance people?

☐ Are old bearings put back after machines are overhauled, or are they discarded? (Modern practice is to throw them away as a preventive maintenance measure, since they represent such a minute part of total equipment cost.)

☐ Do your maintenance men rebuild worn parts, or is the work farmed out to the nearest machine tool builder? (Farming out is frequently cheaper, because tool builders have the latest rebuilding and hardening equipment.

☐ Do you make frequent, unannounced visits to your plants and repair shops?

☐ Have you considered the pros and cons of leased maintenance?

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Shot Peening in Relation to Gear Tooth Scoring

by J. C. Straub

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Shot peening is recognized as a means of increasing fatigue strength of machine parts. Long accepted as a means of correcting fatigue failures, the possibilities of its use in the design of machine parts generally appear to be underestimated. Although used primarily for increasing fatigue strength, if incorporated into gear design, shot peening can be used to avoid other types of failure. Scoring resistance may be increased by use of a finer pitch, but at the expense of bending strength. Shot peening may provide the necessary fatigue strength to accomplish this end.

Test results have indicated an increase of as much as 50 per cent or more in allowable stress in applications requiring a long service life.

The object of this article is to discuss the possible influence of shot peening on scoring tendencies of gears. Directly, shot peening appears to have little influence on the scoring tendencies of a given physical pair of gears. However, it is suggested that the increase in fatigue strength resulting from shot peening can permit a change in the tooth design to favor scoring resistance.

TYPES OF FAILURE IN GEARS

Gear tooth failure can be classified into three major types:

1. Bending failure, or tooth breakage. This is a fatigue type of failure caused by high bending stress in the root fillet. It is the most serious type because a broken tooth usually renders the gears inoperative.
2. Pitting. This is also a fatigue type of failure but occurs on the working profile of the tooth due to high contact stresses. It is not as serious as bending failure because although a moderate noise may develop after the initial stages, the gears can usually be run for an extended period before complete failure occurs.
3. Scoring. This is not a fatigue failure, and may occur early in the life of the gears. It is a welding of the tooth surfaces due to high contact stresses in combination with high sliding velocity. It is characterized by radial scratches on the profiles, somewhat similar to the seizing of a journal bearing.

Presented at the Annual Meeting of the American Society of Lubrication Engineers held in Cleveland, Ohio, April, 1958.

RELATIONSHIP OF DESIGN FACTORS IN THE THREE TYPES

For effective utilization of material in the design of gears, all three types of failure should be considered. If the power is transmitted at a high torque and low speed, scoring resistance may not be a problem. On the other hand, if the power is to be transmitted at high speed, scoring resistance may be an important factor in the design. However, if size and weight of the transmission are to be reduced, it becomes more and more important to consider all three types of failure and to achieve a balance in the design. Resistance to bending failure depends at least in part on the tooth thickness. In contrast, both pitting and scoring are the result of a surface condition and are not influenced directly by tooth thickness. The latter types of failure are strongly influenced by the radii of curvature of the tooth profiles in the area of contact, since contact stress increases with decreasing radii of curvature.

During World War II, a PVT formula for computing scoring resistance was developed on the basis of a large number of tests on carburized and hardened spur gears in aircraft engines, lubricated with mineral oil (1).

More recently, a calculation of "flash temperature" has been used for determining scoring resistance (2) which appears to agree with test results on tractor, aircraft and marine gears (3). These methods differ in their approach to the problem, but in either case the calculation involves the radii of curvature and the sliding velocity of the teeth.

It is quite evident that an excellent tooth design of itself will not guarantee satisfactory performance of a pair of gears. Other factors must be carefully consid-

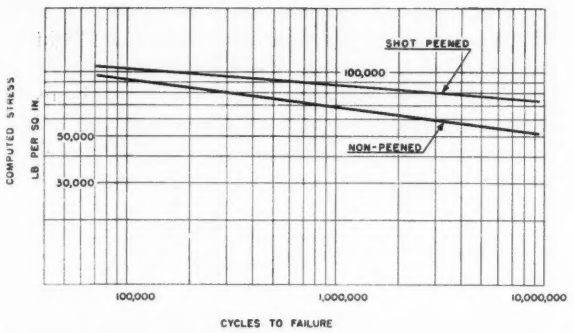

Fig. 1. Fatigue chart of carburized automotive type spur and helical gears, shot peened and non-peened.

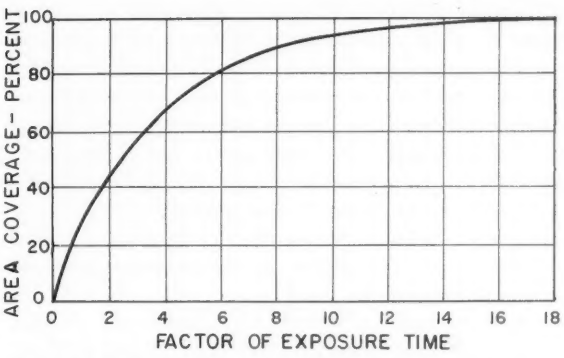

Fig. 2. Relationship between area coverage and exposure time.

ered in the light of resistance to all types of failure. The capacity for trouble-free performance of a pair of gears will depend also upon the accuracy of manufacture, material, heat treatment, mountings, type of lubricant and method of applying it to the teeth, provision for dissipation of heat and other factors.

In some of the elements of gear tooth design, there is no conflict between scoring resistance and bending strength. For example, in a pair of gears with a high ratio, scoring resistance can be improved by the use of long and short addendums. Such proportions may permit adequate tooth action and at the same time avoid contact in the region where contact stress and sliding velocity would be high. This procedure is also advantageous for bending strength because such proportions will normally result in a proportionately greater thickness of the pinion tooth, which will be required to undergo a greater number of stress cycles than the gear tooth.

This "compatibility" does not occur in all of the elements of the design of the teeth. For example, a coarse pitch may be used where high bending strength is required, because of its advantage in a greater tooth thickness. However, this means greater tooth heights for an adequate number of teeth in contact and results in tooth action at a greater distance from the pitch point. Consequently, since the sliding velocity varies directly with this distance, greater sliding velocity is encountered, as well as the likelihood of high contact stresses.

The above factors suggest a fine pitch for scoring resistance. But this would lead to low resistance to bending failure. Therefore, if the bending strength can be increased by shot peening it would permit the use of a finer pitch for greater scoring resistance.

HOW MUCH GAIN IN FATIGUE STRENGTH BY SHOT PEENING

If an increase in fatigue strength can provide a design tool permitting its use for the prevention of scoring, the next question is—to what extent can fatigue strength be increased by shot peening? For such data to be useful in design, it is necessary to consider the increase in allowable stress for a given life. For indefinitely long life, an increase of 50 per cent or more in the endurance limit stress is not unusual.

For shorter life requirements involving higher stresses, the gain in endurance limit stress is more moderate. This is illustrated by Fig. 1 which shows an SN diagram plotted on a log-log scale for shot peened and non-peened spur and helical gears. The gears from which this chart was derived were of the automotive type, carburized and heat treated to approximately 60 Rockwell C. The stresses are computed on the basis of a method developed for automotive gears (4). Each line on the chart represents the average life of a large number of gears. Note that the two lines diverge toward the right. For example, for an expected average life of 100,000 cycles the allowable stress is increased by approximately 10 per cent. For a life requirement of 1,000,000 cycles the increase in allowable stress is somewhat more than 25 per cent. For a life requirement of 10,000,000 cycles, the increase in allowable stress is almost 50 per cent.

The chart in Fig. 1 does not necessarily represent the optimum results which can be achieved by shot peening. This will be further discussed in a later section.

A specific example of the extent to which shot peening can be used to permit the designer to use a fine pitch is one which was recently applied in automotive gearing. In this particular case, a finer pitch was desired by the manufacturer for the purpose of improving noise characteristics. Shot peening permitted a change from 10 pitch to 14 pitch in production, using the same center distance and face width. The manufacturer stated that, in his opinion, this change would not have been possible without shot peening.

APPLICATION OF SHOT PEENING TO GEARS

As in other machine parts, shot peening of gears should be the last operation insofar as the highly stressed areas are concerned. Although the effect of the process is to impose a residual compressive stress on the surface by virtue of a lateral expansion of the surface layer, there is little if any change in the tooth profile, except for an increase in the surface roughness.

In hardened gears, this increase in roughness is very slight. A great number of gears are being peened in regular production today, and the majority of manufacturers makes no attempt to reduce the surface roughness after shot peening. In some cases however, in very large gears, the teeth are cut with a protuberance hob to provide some undercut. The gears are then carburized, heat treated, shot peened, and ground. Thus, the undercut allows the tooth profiles to be ground without disturbing the shot peened surface at the root of the tooth.

Shot peening is effective in increasing fatigue strength of both hardened gears (60 Rc) and gears of machinable hardness.

If it seems necessary to achieve a high degree of surface finish on the working profiles, this can be accomplished on gears of machinable hardness by shaving after peening. Gears of high hardness can be honed (5). Both of these procedures permit the profiles to be processed to the required finish without disturbing the peened surface at the roots of the teeth.

CONTROL OF SHOT PEENING

Since shot peening utilizes the same general type of equipment as that for blast cleaning, one is often confused with the other. Probably the greatest distinction between the two processes lies in the method of inspection. In a blast cleaning operation, the quality of the job is appraised by the cleanliness of the parts after processing. But since the goal of shot peening is an increase in fatigue strength, it will be apparent that visual inspection is not likely to reveal the quality of the peening job, except for the determination of whether the required areas have been struck by the blast.

Control of the peening operation consists in a sense of appraising the blast itself. For this purpose standard Almen test strips and equipment have been standardized by the SAE (6). A thin strip of spring steel is fastened with screws to a solid block and passed through the blast, simulating the work. The lateral expansion of the surface results in a curvature of the test strip when it is removed from the block, the peened side being convex. The degree of curvature is then measured as arc height on a standard cordal distance. A thin strip "A" is used for light impact, and a thicker strip "C" is used for heavy impact. The arc height is designated in thousandths of an inch with the suffix "A" or "C" depending upon the strip used. Thus ".014 A" designates .014" arc height using an "A" strip, as measured on a standard Almen guage.

Specifications for a peening job are based on arc height and some measure of coverage. In establishing such specifications, the arc height should be in keeping with the thickness of the part in the location of maximum stress, as well as the overall requirement of a particular job. It is not practical to generalize on the ideal arc height for all applications because of the wide variation in requirements. Wherever possible, it is good practice to establish by fatigue tests the most effective arc height for a particular job in the light of the overall requirements. The following tabulation is based on a large number of fatigue tests, and can be used as a guide in establishing appropriate arc height for a given thickness in the parts to be shot peened.

Thickness of Part	*Arc Height*
1/16 inch	.004 A
1/8 inch	.008 A
1/4 inch	.014 A
3/8 inch	.018 A
1/2 inch	.021 A
5/8 inch	.007 C
3/4 inch	.008 C
7/8 inch or greater	.010 C or greater

Having set up the specifications it is important, for best results at lowest cost, to obtain the required arc height with uniformly sized shot. Fatigue tests have indicated that, if a high percentage of undersized shot is permitted to accumulate in the peening machine, the expected fatigue strength may not be achieved, even though specified arc height is obtained. For this reason, a peening machine should be equipped with a separator capable of removing broken or undersized shot in continuous operation as soon as possible after it becomes smaller than the original size. Obviously, it is necessary to provide for uniform replacement of the broken shot with new shot by means of an adding device.

In many cases where there is blast cleaning equipment in the same plant, it is quite practical to use this undersized shot in the cleaning operation.

For efficient operation the shot should strike the work at a reasonably uniform angle of impact. If, for example, a large portion of the shot strikes the work at right angles, then any shot which strikes the same area at an angle appreciably less than 90° is ineffective in the same way as undersized shot. This does not mean that the shot must strike at right angles, but rather that the angle of impact should be uniform.

COVERAGE

Coverage in shot peening is related to the degree of exposure to the blast. It may be defined as the proportion of a given area of the work which has received impact of the peening blast.

In any peening operation with constant conditions, there is a definite relationship between coverage and exposure time, as shown in Fig. 2 (7). The horizontal scale is relative and can be expressed in terms of time, or number of passes under given conditions.

Note that coverage approaches 100 per cent as the exposure time is increased indefinitely. For this reason, it is convenient to refer to 98 per cent as a finite measure of approximately "full coverage."

FATIGUE STRENGTH AS INFLUENCED BY COVERAGE

Fatigue tests have indicated that, provided the arc

height is not excessive for the thickness of the part, the fatigue strength increases with increased coverage. In many cases it increases gradually, even after the exposure time far exceeds that required to obtain 98 per cent. This appears to be especially true in peening parts of irregular shape such as gear teeth. For example, gears peened with an exposure of three times that required to obtain 98 per cent coverage give results well above the chart for peened gears in Fig. 1. This is referred to as a coverage of 3. Gears peened with a coverage of 6 to 8 show even greater increase.

In order to utilize shot peening to best advantage, it is important to select a degree of coverage which is best suited to the requirements of the particular job. If shot peening is incorporated in the design of gears, it is likely that the purpose is to reduce the cost of production of such gears (8). Such being the case, it is only natural to reduce costs to the practical minimum. It is important, to specify a degree of coverage in accordance with the economics of the gears in question. For example, with gears involving a very high production rate, the cost of manufacturing the gears may be relatively small. In such a case, it might be economical to specify a coverage in the neighborhood of 98 per cent, or possibly even less. The fatigue tests mentioned above have shown a more gradual increase in average life after a coverage of about 80 per cent is obtained. Reference to Fig. 2 will reveal that the exposure time required to obtain 80 per cent coverage is less than half that required for 98 per cent. Consequently the additional increase in fatigue strength at full coverage may, in some cases, not be warranted. It should be mentioned, however, that where high production rates are involved, the capacity of a wheel type machine may be so high that the conveyor speed through the peening machine is limited only by the capacity for handling the gears to and from the machine. A coverage of 3 is not uncommon in high volume production of relatively low cost gears.

There are applications in which the cost of manufacturing the gears is sufficiently high that the cost of shot peening is insignificant, even though a coverage of 6 or 8 is specified. In this case, a much greater increase in fatigue strength can be obtained at negligible cost.

CHOICE OF SHOT SIZE

The size of shot best suited for a given application depends upon the arc height and coverage to be obtained.

For steel gears, it has been the author's experience that a small shot at high velocity will result in about the same increase in fatigue strength as a larger shot at correspondingly lower velocity, provided the same arc height and coverage are used. However, as mentioned earlier, whatever shot size is used, it is important to maintain uniformity of size for efficient operation.

Since the number of shot particles per pound varies inversely as the cube of their diameter, it follows that coverage is obtained much faster with smaller shot. This of course means that the shot velocity must be higher to obtain the same arc height. However, in spite of the fact that shot usage per hour increases considerably with shot velocity, the increased production compensates for it, so that the shot usage per part peened is about the same. The end result is that the most economical operation can usually be obtained by using the smallest uniformly sized shot with which the required arc height can be obtained.

TYPE OF SHOT

As in the case of selection of shot size, the type of shot should be selected on the basis of lowest operating cost for efficient operation. It has been demonstrated (7) that it is not necessary that shot for peening gears with a hardness of 60 Rc have the same hardness as the work. Equally good results have been obtained by peening fully hardened work with cast steel shot (42-50 Rc) and with chilled iron shot (60 Rc). A good grade of cast steel shot has a far greater resistance to breakage and therefore is much more economical. In addition, the maintenance cost of a peening machine is much less with steel shot because of the lack of sharp edges which are prevalent in chilled iron shot as it breaks down.

Cut wire shot can also be used if its hardness is sufficiently high and uniform. However, its use is usually limited to the larger sizes because of the increase in cost per pound as the size decreases. This conflicts with the economy of using smaller sizes for faster coverage.

When steel shot is used for peening it is important that new shot be run in for a short period before peening the parts. This is because of the work hardening of the shot itself, and is evidenced by an increase in arc height after new shot has been run in. This pertains only to starting a machine with all new shot. After the machine has been in operation for a few hours, the amount of new shot for make-up is a negligible percentage of the shot in circulation.

EQUIPMENT FOR SHOT PEENING

To insure an adequate peening job, the work should pass through the blast in a mechanically controlled cycle, so that all critical areas will be uniformly covered.

The equipment for accelerating shot is generally of two types—one which utilizes compressed air and the other a rotating bladed wheel. Air equipment may be of the induction type which accelerates the shot within the nozzle itself, or direct pressure type in which the shot is added to the air stream under pressure.

The induction nozzle is sometimes used in applications which require very few small parts to be peened. The direct pressure type is capable of throwing more shot at a higher velocity and therefore can be used in cases where more parts are required. In applications involving a reduction of cost by shot peening in design,

(*Cont. on page 119*)

Determining Wear in Automotive Engines

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Several laboratory test techniques are used for determining wear in both single-cylinder and multi-cylinder engines and are applicable to specific wear problems affecting lubricant development. Piston ring, piston pin, and valve train wear under various operational conditions are discussed in this study.

An automotive engine is a machine which burns fuel and produces useful mechanical work. Its various rubbing surfaces are lubricated with oil to reduce friction and wear. Changes in design and performance requirements associated with the so-called "horsepower race" in the past few years have accentuated the wear problems in engines and stressed the need for developing better lubricants to prolong the useful life of these engines.

Modern engines have presented a variety of critical wear problems. Corrosive, abrasive and frictional wear of piston rings and bearings has been described in the literature (1, 2, 3, 4, 5). Valve train wear is a major problem (6, 8, 10). Changes in modern driving habits, including more stop and go driving and high speed toll road operations, also adversely affect engine component wear.

Wear measurements on engines are made to predict the service performance or useful life of the engine or of some particular component part which experience shows to be critical with respect to wear and/or failure. There are two popular methods of measuring wear at present (a) the direct physical measurement (micrometry and gravimetry) of worn parts, and (b) the indirect measurement of radioactive material worn off an irradiated engine component. Chemical analyses for metal appearing in the used lubricating oil should also be considered as a third method, but it is difficult to ascertain which engine parts are showing the most wear by this procedure.

The development of higher performance lubricating oils for modern engines has necessitated the development of better wear tests. This article will describe various test techniques used in determining wear and their applicability to specific wear problems affecting lubricant development.

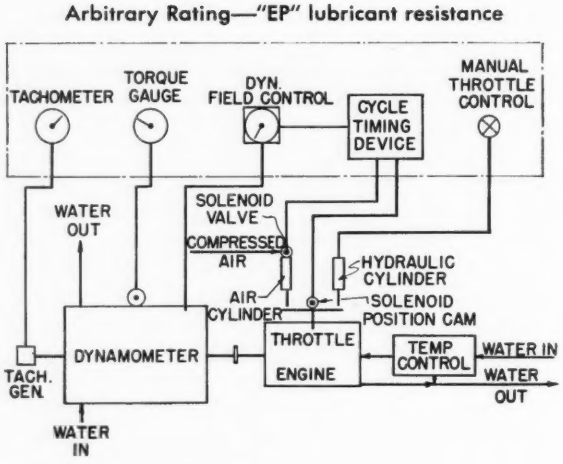

Fig. 1. Laboratory engine cycling test apparatus.

PHYSICAL METHODS

During the past few years, a number of engine manufacturers and oil companies have reported field service problems arising from excessive wear of "upper level" engine parts—piston rings, piston wrist pins, rocker arm shafts, tappets, and cams. Prior to this time, wear of these parts was not a serious problem and engine test procedures were not specifically designed to evaluate wear.

Preliminary test development work during this period indicated that test work with critical automotive engines was more feasible than trying to impose severe operational conditions on the standard laboratory engines used for other types of test work. Consequently, dynamometer test stands were equipped with current model automotive engines, both V-8 and

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6 cylinder models, and test conditions were adjusted to simulate actual driving conditions.

A schematic drawing of the main features of a typical laboratory engine test set-up is shown in Fig. 1. The engine and dynamometer are operated from a remote control panel board outside the test cell. Manual and automatic cycling controls governing throttle position, dynamometer loading, and jacket temperatures are used to simulate idling, acceleration, deceleration, and wide open throttle driving conditions for passenger car operation. More elaborate cycling controls, such as magnetic tape records of actual road operation, are being used in some laboratories.

WRIST PIN WEAR

A typical wear problem occurred in an early high output V-8 overhead valve engine. This engine was experiencing audible piston pin noise and excessive wear at fairly low mileages in field service. The piston pins were made of hardened steel and mated to aluminum alloy piston bosses. Appendix I lists the details of a laboratory 500-hour cycling test procedure which was finally developed to duplicate the excessive pin wear of field engines. The 500-hour laboratory test is equivalent to approximately 25,000 miles of road operation at speeds up to 80 mph.

Fig. 2 shows the results of a series of tests run by this procedure in the development of a lubricating oil "fix" for this problem. Oil *A* was a high quality service station lubricant which ran only 103 hours before audible wrist pin noise occurred. Additional anti-oxidant and corrosion inhibitors of the zinc dithiophosphate type were found to decrease wear as typified by Oil *B*, but did not increase pin life significantly. Different types of detergent additives—Oils *C* and *D*—did not help. An oiliness agent finally proved to be satisfactory as shown for Oils *E*, *F*, and *G* which contained increasing percentages of the additive.

Development of this lubricating oil "fix" is an example of the laboratory work required to solve various engine component wear problems. The wrist pin "failures" which caused audible knocking were not easy to determine, even in the laboratory with a stethoscope. The arbitrary definition of failure after 0.0009 in. diametral wear was established after several hundred hours of testing. The 500-hour tests used for reliable physical measurements of wear require considerable time and manpower for a comprehensive evaluation of such lubricants.

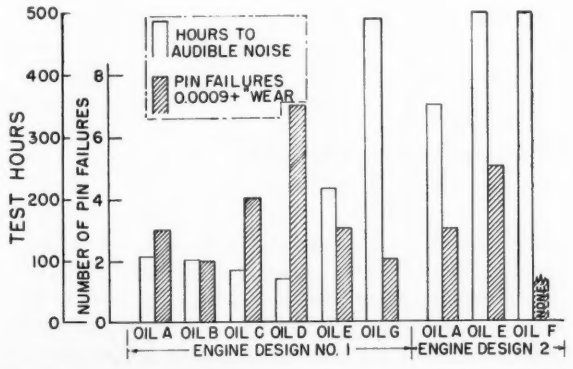

Fig. 2. V-8 engine piston pin life.

Subsequent modification of the oil grooves in the piston boss in new model engines also alleviated this problem as shown when Oils *A* and *E* were retested in the newer engine. The mechanical "fix" actually accounted for a 300 per cent increase in service life with Oil *A* before audible pin noise occurred due to an overall decrease in maximum pin wear. The interrelationship between mechanical and lubrication factors in wear problems cannot be over-emphasized.

VALVE TRAIN WEAR

The contact areas between the cam lobe and tappet and between the rocker arm shaft and rocker arm in some current engines have also been shown to be sensitive to differences in lubricants, although equally dependent upon proper design and metallurgy. A great deal of information (6, 8, 10) has been published on this subject, including a recent comprehensive field investigation of valve train wear problems by the CLR Camshaft and Valve-Tappet Wear Group.

Early work on cam and lifter wear was done with the LS-5 and LS-6 test procedures developed by Bidwell et al (6). The LS-5 test is run in a 1953 six-cylinder valve-in-head engine and the LS-6 test used the 1954 model of the same engine. In the LS-5 test, chilled iron lifters are run with a forged steel camshaft. The LS-6 test is run with six carburized steel lifters and six hardened alloy iron lifters on a hardened alloy iron camshaft. Valve spring loads are the most critical modification of these tests. The 80-pound spring overload is intended to maintain a stress magnitude between the cam and lifter faces approximating that of severe low speed engine operation while running the engine at a higher speed to get more contact cycles per test.

These tests are not in general use today since the heavy spring loading is not representative of normal service and field correlation has not been satisfactory. The lifter spalling or pitting phenomenon recorded in the LS-5 test is primarily due to metal fatigue caused by repeated stressing of the lifter surface material. If this surface layer wears at a low uniform rate, none of the metal particles will be cyclically stressed enough times to produce fatigue cracks and subsequent spalling before being worn away. The abnormal spring loads of the LS-5 test tend to promote spalling and erratic wear of this particular metal combination. Lifter rotation has been found to be beneficial in reducing spalling tendencies by alleviating the "hammer" effect on a given area of the lifter. The alloy cast iron parts in the LS-6 test usually show little wear and are now used extensively in the automotive industry.

Recent work on valve train wear has been done with the cycling type endurance test procedure dis-

cussed previously. Slightly higher loads are incorporated in the basic 500 hour cycling schedule as newer engines with higher power outputs are tested. In addition, a dummy engine test has been developed for screening purposes. It is one developed by an engine manufacturer and consists of a V-8 engine block and heads with a special straight crankshaft driven by means of an electric motor. The engine block is completely assembled with the exception of pistons and exterior equipment. The 100 hour test is run at a constant speed of 1500 rpm. without crankcase oil temperature control. Details of this procedure are given in Appendix II.

Fig. 3 shows a comparison of wear on camshafts and tappets obtained by these two test procedures. Oil *I* is a commercial multi-graded motor oil and Oil *J* is a similar formulation with an experimental anti-wear additive of the dithiophosphate type. Both maximum and average wear data for the rubbing surfaces are shown. The anti-wear agent was more effective in reducing wear of the cam lobes than of the lifters. Although the wear figures for the 500 hour "fired" test are generally higher than the 100 hour dummy test, the wear trends are similar. The dummy engine has proven to be a satisfactory screening tool even though consistently less severe than the 500 hour cycling test or a 100 hour "fired" test under the same constant speed operating conditions of the dummy test.

Wear of rocker arm shafts and rocker arms must also be considered in motor oil evaluation. Excessive wear on these parts can have adverse effects on valve clearances with subsequent valve and lifter failures. Wear of these engine components is also evaluated by the procedures described above for the cams and lifters in the same engine tests.

Fig. 4 shows wear of this type in both dummy and fired engines. Oils *I* and *J* are the same as those shown in Fig. 3, with Oil *J* being quite effective in reducing wear of these parts. This graph also illustrates the effects of mechanical and metallurgical changes on the wear problem. The "new design" label on the second bar of each pair indicates a design modification for a later model of the same V-8 engine. This change consisted of a phosphate coating on the rocker arm shaft and different oil groove geometry in the rocker arm bores. The mechanical "fix" alone decreased the average wear on both oils nearly 60 per cent in the fired engine tests, but was not quite as effective in the dummy engine tests. Rocker arm shaft wear in the dummy engine is normally not as severe as that of the fired engine. This phenomenon is believed to be related to some effects of combustion shock or vibration and combustion product contamination of the oil in a fired engine which are not present in the dummy engine tests.

Fig. 5 compares rocker arm shaft surface finishes in the contact areas between rocker arms and the shaft for the same series of tests. This evaluation is made to determine the type of surface protection afforded by

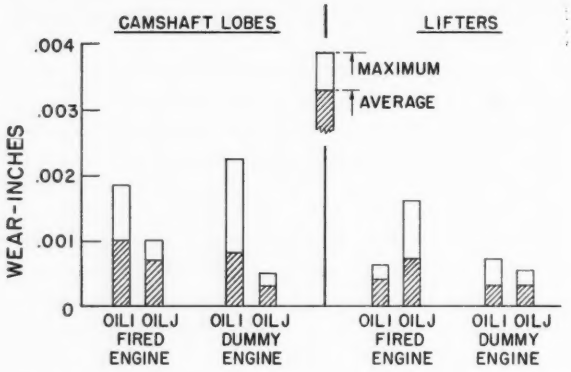

Fig. 3. Engine valve train wear.

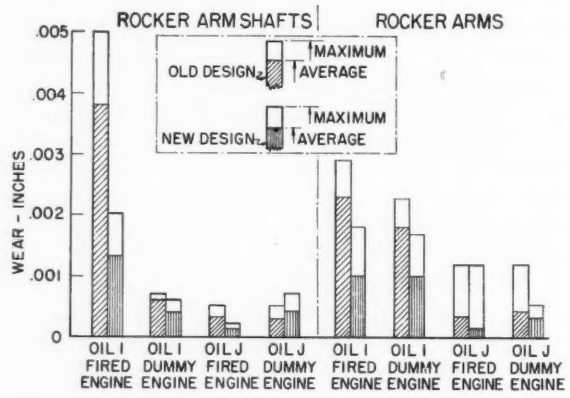

Fig. 4. Engine valve train wear.

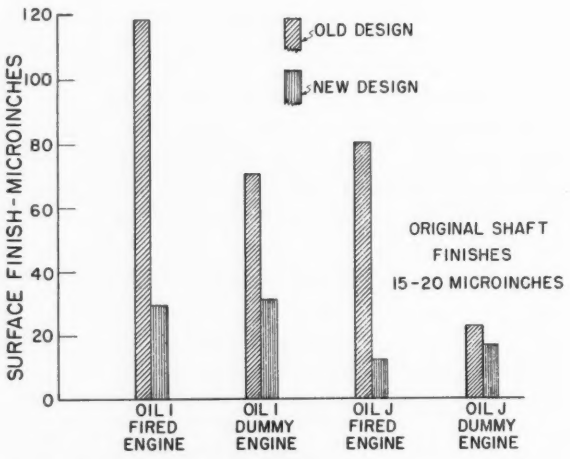

Fig. 5. Rocker arm shaft surface finishes.

different anti-wear agents. A coarse or rough surface is indicative of metal-to-metal contact under boundary lubrication conditions and should be susceptible to EP additive treatment. These data show that the phosphate coating on the new design rocker arm shaft fulfills this function since the experimental Oil *J* was not

as effective in this respect as on the old design parts. Surface finish data correlation between the dummy and fired engine procedures is similar to that for actual wear measurements.

RADIOACTIVE METHODS

The conventional direct physical methods of measuring wear described in the preceding portion of this paper require considerable amounts of time and manpower, and in some cases lack sufficient accuracy. Radioactive screening tests shorten test time and increase accuracy, and provide important data, not easily obtained in any other way, for the laboratory and field evaluation of automotive lubricants. However, radioactive methods cannot replace all of the conventional tests since such factors as oil deterioration, deposits, etc., are also rated in the 500 hour endurance tests and must be included in the overall evaluation of modern motor oils.

Measurements of engine wear by use of radioisotopes has attracted considerable interest in the past few years (3, 4, 7, 9, 11), largely because such methods are extremely sensitive and changes in wear rates occuring in a few minutes or hours can be readily measured. The effects of a large number of operating variables can be evaluated in a short period of time as contrasted with direct physical methods. The effects of transitory phenomena—oil changes, cold starts, fuel/air ratio changes, different fuels, etc.—can be observed which would be almost impossible to detect by any other method. In addition, mechanical adjustment factors need not be major test variables since the engine is not dismantled after every test to make radioactive wear measurements.

The use of radioisotopes to measure engine wear does require special instrumentation and safety precautions for handling the "hot" engine and oils. The bulk of the work in this field has been done with radioactive piston rings, although studies of other irradiated engine parts are also under active consideration. Piston ring wear has been studied in a single-cylinder Cooperative Oil Test engine (C.O.T. engine) in order to reduce the number of operating variables inherent in a multi-cylinder engine. Scintillation detection equipment is used to measure radiation from wear particles because of its greater sensitivity and longer life than geiger counting equipment. This equipment in turn has eased personnel safety problems by allowing the use of lower irradiation levels or activity for the piston rings or longer useful ring life in the engine prior to overhaul. Irradiation of each cast iron ring to an activity level of approximately 5 millicuries is done at the Argonne National Laboratory.

The operation of a "hot" engine is quite similar to that of a conventional engine. As the ring wears, the radioactive particles are carried to the sump by the lubricant on the cylinder wall. The oil, carrying suspended wear particles, is pumped from the engine sump through a cooler into a chamber surrounding the scintillation counter crystal. After leaving the counter, the oil is returned through a reheater and back into the engine. The wear rate, in terms of radiation pulses or counts per minute from the radioactive wear particles, is continuously measured by means of a ratemeter connected to the scintillation counter, and recorded. The engine and dynamometer are normally operated from a remote control panel board outside the test cell. A schematic drawing of this test apparatus is shown in Fig. 6.

To convert "counts per minute" data to actual milligrams of wear, a known weight of irradiated ring material is dissolved in hydrochloric acid which is then diluted to a known volume. The ring material is obtained from filings taken from the inner periphery of the irradiated ring. This solution of radioactive iron is then placed in the chamber surrounding the scintillation crystal and counted. Since the concentration of iron in the solution is known, the concentration of wear particles in a similar quantity of test oil can be determined.

Owing to the relatively short half-life of the iron 59 isotope (47 days), it is necessary to correct all data to a common base or reference line. This is done by taking filings from an irradiated ring and measuring or counting their activity at weekly intervals. Decay correction factors are then determined by plots of these data on semilog paper.

STEADY STATE ENGINE WEAR

Details of two typical test procedures for the Cooperative Oil Test engine are given in Appendix III. The first of these methods is intended to simulate frictional wear conditions or over-the-road operations. The second method typifies corrosive wear or low jacket temperature stop-and-go operations. The latter test conditions are quite similar to those of the Chevrolet FL-2 test procedure and are considered to be important in the evaluation of current motor oils. For oil studies, a standard laboratory reference fuel was adopted for all test work to eliminate any test variables due to seasonal variations in the composition of commercial fuels.

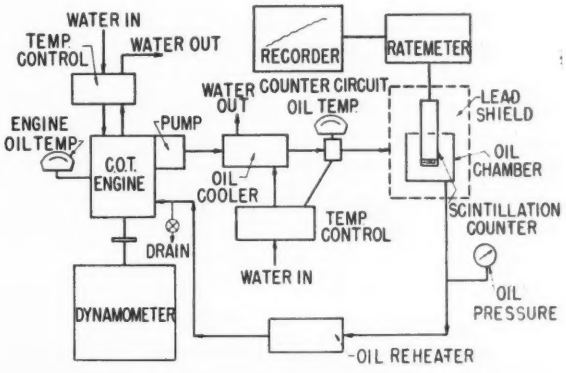

Fig. 6. Radioactive piston ring test apparatus.

Piston ring wear data for two motor oils are shown in Fig. 7 to illustrate the ability of these two test procedures to separate oils on the basis of their performance under corrosive and non-corrosive operating conditions. Oil *L* is a typical "MM service" lubricant (SAE 20W grade) while Oil *K* is a heavy duty "MS-DG service" multi-graded oil (SAE 10W-30 grade). A study of these curves also indicates that two phases of wear are involved; namely, start-up wear and wear under stabilized or steady state operating conditions. It will be noted that the wear rate is essentially constant after the initial one-half hour warm-up or start-up period.

The data in Fig. 7 show that the wear rate for Oil *K* is only one-third that of Oil *L* under these operating conditions, while the wear rates for the less severe frictional wear test conditions are nearly equal for the two oils. These data are based on the constant wear rate portions of the curves. More work has been done with the corrosive type wear test since these test conditions give higher wear rates and more clearly show differences between oils.

The influence of engine temperatures on piston ring wear is illustrated in Fig. 8 for an experimental Oil *H* (SAE 10W-30 grade), for its base stock alone, and for a viscosity index improver additive. The beneficial effect of the viscosity index improver added to the base stock under such operating conditions is shown here as an example of what can be done with individual additive components using the radio-active technique. These data indicate that cylinder wall temperatures below 200F are conducive to corrosive wear while those above 200F are more indicative of low level frictional wear. An upper cylinder wall temperature range of 160-165F appears to be suitable for evaluating motor oils and their additive components. These temperatures are the minimum controllable in this test set-up with jacket water in the 55-65F range without auxiliary jacket cooling.

Fig. 9 shows the effects of viscosity and different additive levels on top ring wear for a series of typical motor oils. These data indicate that the lighter grade lubricants are somewhat better under corrosive wear test conditions, although the 5W-20 oil is borderline in this respect. Additive compounding levels are not too critical with respect to ring wear, but all of the compounded oils investigated are better than a straight mineral oil.

Top compression ring wear rates for the individual additive components of a typical multi-graded motor oil are shown in Fig. 10. It will be noted that the viscosity index improver and detergent type additives are especially effective anti-wear agents under these corrosive type operating conditions.

As stated previously, the effect of many operating variables on wear can be studied with radioactive tracer techniques. Fig. 11 illustrates the effects of carburetor flooding and abrasive material on top ring wear in two different tests with Oil *N* (SAE 20W grade)

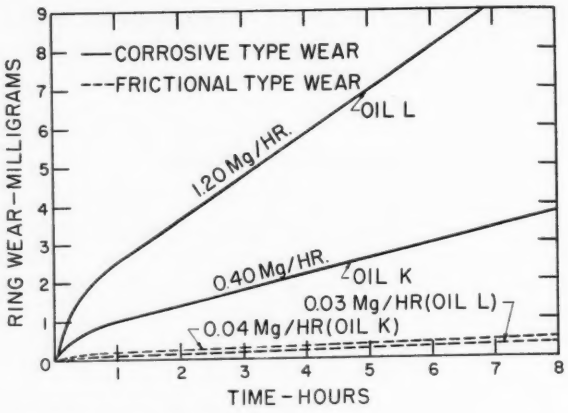

Fig. 7. Top compression piston ring wear.

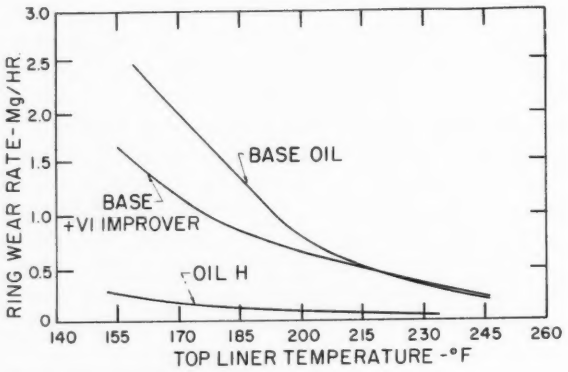

Fig. 8. Top compression piston ring wear.

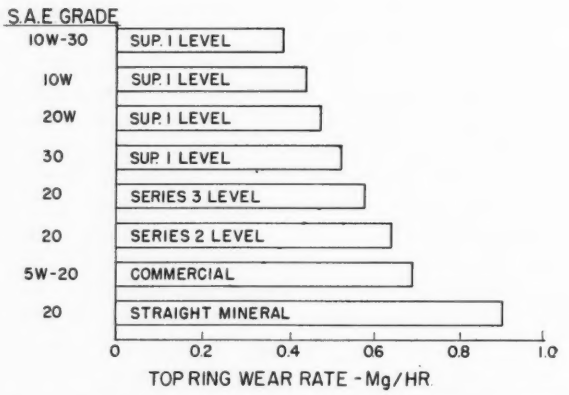

Fig. 9. Effect of oil additives and oil viscosity on corrosive ring wear.

under frictional wear operating conditions. Momentary flooding such as that caused by a sticking choke will greatly increase wear for a short period of time. Temporary interference with the lubricating oil film on the cylinder wall allows metal-to-metal contact with resultant accelerated wear until the oil film is restored.

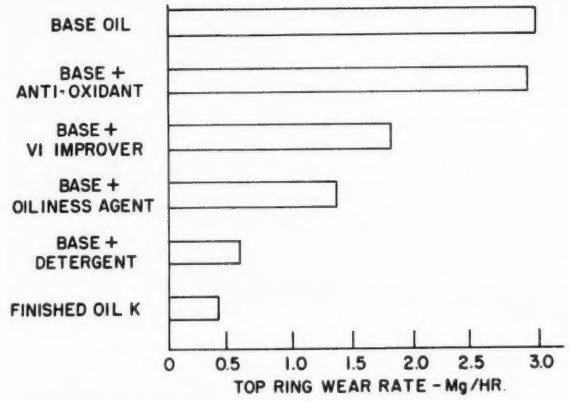

Fig. 10. Effect of motor oil additive components on corrosive ring wear.

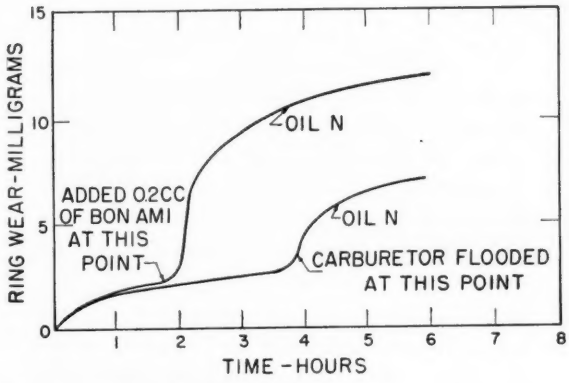

Fig. 11. Top compression piston ring wear.

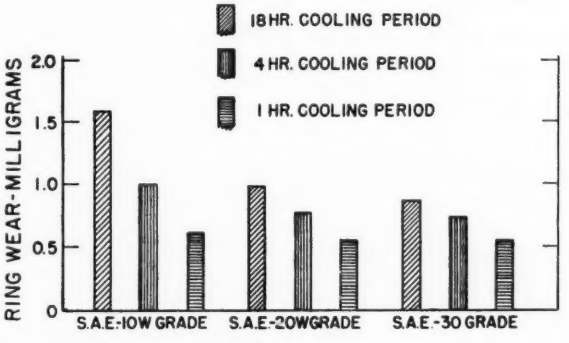

Fig. 12. Effect of engine down-time on start-up wear.

Abrasive material entering through the carburetor air intake also accelerates wear until the contaminating material is washed away.

START-UP ENGINE WEAR

As mentioned earlier, two phases of wear are involved in these laboratory engine tests. Wear rates under steady state operating conditions have been found to be reliable indicators for classifying lubricants, but the initial start-up wear is also important. Significant piston ring wear has been shown to occur in the first few minutes of cold engine operation and during stop-start city driving. Increasing speeds and load and quick accelerations also increase ring wear in passenger car engines (11).

Fig. 12 shows some effects of engine down-time or cooling periods on start-up wear. Details of a carefully controlled engine starting procedure used in this study are given in Appendix III. Each bar represents total top ring wear for 30 minutes of operation after a specified engine cooling period of one, four, or 18 hours at room temperature. The three test oils are similar in composition to the Supplement I oils shown in Fig. 9.

These data indicate that the three viscosity grades of oil are approximately equal in anti-wear characteristics after a moderate one-hour engine cooling period. However, start-up wear for the SAE 10W grade oil is approximately 30 per cent greater than that for the SAE 20W and 30 grade oils after a four hour cooling period and 70 per cent greater after an 18-hour period. Start-up wear for the SAE 20W grade oil is also somewhat higher than that for the SAE 30 grade oil after the longer cooling periods, although not as significant. The heavier grade oils apparently afford better protection against corrosion caused by condensation of combustion products on the piston rings and resultant start-up wear in a relatively cold engine.

Start-up wear should also be compared with the "steady state" wear rates shown in Fig. 9. These data showed that lighter grade lubricants gave somewhat less wear following the initial engine starting and warm-up period. However, it is important to note that over 25 per cent of the total ring wear in a six-hour test occurs during the first 30 minutes of operation of a cold engine.

FUTURE

Major wear problems occurring in modern automotive engines and several laboratory test techniques used in determining wear relative to lubricant improvement have been described in this paper. More work remains to be done on methods for evaluating wear since similar problems can be expected in future high output engines.

Short term screening tests must be developed to provide reliable wear data on the various components or additive combinations of lubricants prior to full scale evaluation. Wear studies of different irradiated engine parts appear to fulfill this requirement. Extended service type engine tests also must be run under realistic road operating conditions as part of the final proof testing of a finished lubricant.

(Cont. on p. 118)

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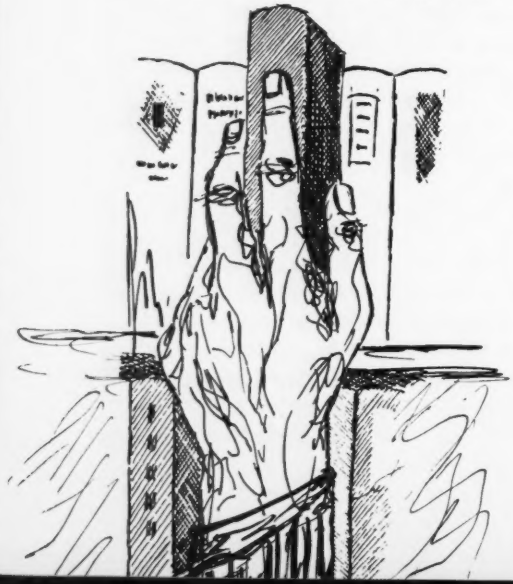

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ASLE Technical Digest

Papers which, because of space limitations, cannot be published in full in Lubrication Engineering or ASLE Transactions.

"Design Consideration in the Circulation and Filtration of Cutting and Grinding Compounds," by I. M. Rosenbaum, Van Straaten Chemical Co., Chicago, Ill.

Design, layout and maintenance of a central coolant system in the handling of cutting and grinding fluids present interesting problems. Basically the purpose is to deliver the coolant at a rate sufficient to cool the tool effectually and to remove the cuttings rapidly and completely from the cutting edge and the work. Metal cutting and grinding oils are precisely compounded; their original chemical nature must be preserved as far as possible, and they must be kept free from contaminants. Rancidity and abrasives such as grinding fines can seriously impair the resultant surface finish.

Arrangement of piping and fittings in the system has a direct relationship to the condition of the fluid being circulated. Since cleanliness is so important there must be adequate provision for removal of swarf, metal chips and entrained dirt; otherwise lines could become obstructed to reduce—or even cause—stoppage of oil flow. Suitable provisions must be made for oil draw-off or flushing from dead ends, or machines which may be out of service for any length of time. Stagnant oil accumulations can breed bacteria and lead to rancidity.

In this case, pressure relates to oil flow. In the main header and down lines, the fluid is under sufficient pressure to exert a nominal flushing effect on foreign matter build-up. The gravity flow returns are a different matter however, necessitating suitable pitching and reduction of fittings, elbows, etc., wherever possible to facilitate the flushing effect in flow-off to the settling tank. Here reconditioning of the fluid is expedited if velocity and agitation are reduced so that the maximum of swarf and chips is settled out to reduce the subsequent load on the filter unit. A good sized settling tank is an important adjunct.

To obtain maximum filtering effect, coolant must be compatible with filter medium. Where this may involve various grades of paper or plastic, or weaves of cloth, the chemical characteristics of the coolant along with the chip-carrying capacity and wettability must be considered. For this reason compatability tests are advisable to obtain the best filtering effect, bearing in mind that any change of either filter medium or coolant should be considered only after the cooling effect with relation to surface finish has been studied. Never use a combination of materials which would lead to removal or absorption of any of the chemical constituents of the coolant, or which would tend to break the emulsion-forming ability. Straight mineral oils present no problem; the soluble oils or compounds, however, require most careful consideration. (*Digested by A. F. Brewer*)

"The Cutting Fluid Problem," by L. V. Colwell, University of Michigan, Ann Arbor, Mich.

Probably the most important objective in the use of a cutting fluid is surface finish. This is arrived at by the cutting action of the tool against the work. The amount of cut, of course, depends upon the degree of reduction. In any case, the life of the tool is a factor. Surface finish can be very sensitive to the cutting conditions. These may vary according to the nature of the work and the type of cut. As far as the cutting fluid is concerned two basic qualities are desired—cooling and lubrication. These must be maintained under varied conditions of pressure, temperature, quantity of heat, rubbing velocity and the attendant forces which may develop. All prevail as the cutting edge of the tool makes contact with the work and the chip is formed.

The importance of lubrication can be realized by noting that approximately one-third of the total work required to cut the metal is needed to overcome friction between the chip and the tool. From another viewpoint, the total cutting energy and therefore the total heat generated could be reduced by more than one-half through perfect lubrication between the chip and tool face. Obviously, it is worthwhile to reduce friction as a means of reducing heat and temperature. But pressure must be considered because the maintenance of lubrication becomes more difficult under increased pressures.

Temperature, in turn, relates to tool wear and also to hardness. Hard steels produce high temperatures at low friction. This suggests the use of cutting fluids of high cooling ability for high strength materials and for products of pronounced lubricating ability for low-strength high friction materials. Where the high temperature zone may be relatively inaccessible for lubrication in continuous cutting, cooling is made available as a protective medium by directing a high velocity stream of fluid to the tool face near the rear edge.

In regard to the relation of surface finish to tool wear, the former can be very sensitive to small amounts of tool wear particularly in relatively thin-chip operations such as reaming, form-turning, gear shaving, broaching and even some gear cutting.

Research studies have suggested the following interesting and valuable conclusions:

1. High temperature cutting operations require dominant cooling.
2. Low-temperature, low-pressure operations enable effective lubrication.
3. Most high temperature operations also include high pressures between chip and tool face which are not conducive to effective lubrication.
4. Coolants should be applied as closely as possible to the shear zone and chip-tool interface.
5. Effective lubrication between the flank of the tool and work often is a critical necessity.
6. Lubrication is the only mechanism whereby cutting fluids can improve surface finish. (*Digested by A. F. Brewer*)

Note: Unabridged copies of the foregoing papers are not available at Society headquarters. Those wishing additional information about these papers should contact the author.

Determining Wear in Automotive Engines

(*Cont. from p. 115*)

Both lubricant and engine manufacturers need to know more about the "how and why" effects of lubricant properties and metallurgical variables on the wear process in an engine.

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APPENDIX I

Laboratory Engine Cycling Wear Test Procedure

I. BREAK-IN CONDITIONS

Engine is operated twenty-four hours at speeds of 1000 to 3600 RPM and loads up to 80 BHP. Jacket water temperature is controlled at 180 ± 5F and crankcase oil temperature is kept below 250F.

II. TEST CONDITIONS

Cycling schedule: (1st 10 hours on automatic control; next four hours on manual control.)

TIME	SPEED, RPM	LOAD, BHP	JACKET WATER, °F IN	JACKET WATER, °F OUT	OIL SUMP, °F
5 minutes	450	To Fall	To Fall	151 ± 2	To Fall,
15 seconds	Accelerate to 2500	150	To Fall	151 ± 2	Do Not
35 seconds	Decelerate	—	To Fall	151 ± 2	Exceed
Repeat above acceleration and deceleration cycle 6 times; then					
20 minutes	1500	12.5	To Fall	151 ± 2	250
Repeat above 30-minute sequence 20 times; then					
4 hours	3500	70+	To Fall	151 ± 2	250

Continue 14-hour cycles to 500 test hours or failure

Fuel—Commercial (premium grade)

III. TEST DATA

Log engine operating data every two hours, check crankcase oil every 14 hours and drain every 28 hours.

Record dimensional changes and condition of all engine parts (before and after test measurements).

APPENDIX II

Dummy Engine Wear Test Procedure

I. ENGINE AND TEST PARTS

V-8 engine cylinder block and heads with a special mainshaft (no piston or rod assemblies) and belt driven by a 5 HP electric motor.

Rocker arms
Rocker arm shafts
Camshaft
Lifters

II. TEST CONDITIONS

Speed	1500 RPM
Load	None
Oil Sump Temperature	170 ± 10F
Test Duration	100 hrs.

III. TEST DATA

Record dimensional changes in valve train parts and surface finish of rocker arm shaft rubbing surfaces.

APPENDIX III

Radioactive Piston Ring Wear Test Procedures

I. TEST APPARATUS

Carbureted Cooperative Oil Test single cylinder engine, 2 3/4 inches × 2 3/4 inches, 16.25 cu. in. displacement with radioactive cast iron top compression piston ring

Auxiliary external oil circuit
Radiation pulse counting instrumentation
Safety monitoring and handling equipment

II. TEST CONDITIONS

	FRICTIONAL WEAR	CORROSIVE WEAR
Speed, rpm	2400	1500
Load, bhp	4.8(97.6 BMEP)	3.0(97.6 BMEP)
Air/Fuel Ratio	12.5–12.7	12.5–12.7
Jacket Temp., °F	212	To Fall
Oil Sump Temp., °F	170	135
Top Liner Temp., °F	To Fall	160–165
Bottom Liner Temp., °F	To Fall	To Fall
Test Duration hr.	6	6
Test Oil Charge	5.2 lb.	5.2 lb.

Fuel—Laboratory reference blend of L-4 and Isooctane

III. TEST DATA

Log engine operating data and use chart recorder in radioactivity counter for continuous record of wear in terms of radiation pulses (CPM).

IV. CONTROLLED "START-UP" WEAR TEST OPERATION

A. Motor engine with dynamometer for one minute at 1250 rpm (no load)
B. Adjust throttle and run engine for one minute at 1500 rpm (no load)
C. Readjust throttle and run engine for 28 minutes under regular corrosive wear test conditions. Do not use heat on oil sump during this period.

Note: Allow engine to cool down normally at room temperature (Approx. 75F) for the 18, four, or one hour idle shutdown periods between test starts.

LUBE COURSE PROGRAM

(*Cont. from p. 94*)

The practical approach to maintenance and lubrication of ball and roller bearings.

Short discussion and distribution of Gulf Oil Corp. manual "Bearings and their Lubrication."

SEALS AND PACKINGS: E. W. Fisher, Major Product Eng. Dept., Garlock Packing Co., Palmyra, N. Y.

A survey of this field covering mechanical seals, gaskets, packings and stuffing boxes, and emphasizing the details of lubrication and maintenance.

PLANT VISITATION

1:30 P. M.-4:30 P.M.

Lackawanna Works of Bethlehem Steel Company. Busses leave Statler Hotel entrance at 1:30 p.m. Guided tour of the Lackawanna Works with emphasis on lubrication applications.

Thursday, April 23, 1959

9:00 A.M.-12:00 Noon

PLANT LUBRICATION & ORGANIZING A PLANT LUBRICATION PROGRAM: Charles L. Pope, Eastman Kodak Co., Rochester, N. Y.

An informal discourse on the subject of lubrication of industrial machinery as it pertains to the purchasing agent, operator, maintenance man and the role of the plant lubrication engineer in organizing a plant lubrication program.

QUESTION AND ANSWER PANEL:

Questions submitted on cards all during the course will be answered at this time.

NO AFTERNOON SESSION

NOTE: Lubrication course students will be permitted to attend any technical session of their choice.

Factors Affecting Hi-Temp Gear Operation

(*Cont. from p. 103*)

and case carburized gears. This band seems to be pretty well established for this combination. The silicone oil when run with case carburized gears, produced about the same scoring resistance as the MIL-L 7808C oil test. The same silicone oil when run with nitrided gears, gave a much improved performance.

Large amounts of valuable information have been gathered from investigations and test stand operations. Plans are underway to continue on the various programs until a more complete understanding of all the factors that affect high temperature gear operation have been obtained.

Actually, the high temperature gear test facility has been a worthwhile and vital link in the progress made to date. The results from running full sized aircraft gears seem to produce extreme realistic results. It is expensive and time-consuming but completely necessary.

Shot Peening in Relation to Gear Tooth Scoring

(*Cont. from p. 109*)

the volume of production is usually such that the natural choice is the wheel type of equipment. This is because of the fact that the wheel type is capable of accelerating many times as much shot, to the same velocity, with an expenditure of a fraction of the horsepower required in the compressed air types.

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